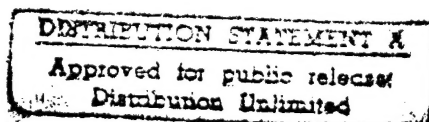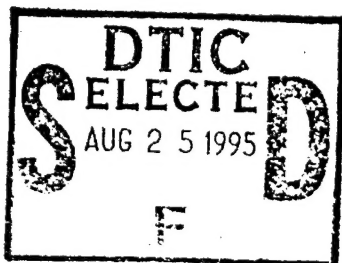# NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 779

THE EFFECT OF INCREASED COOLING SURFACE ON PERFORMANCE OF AIRCRAFT-ENGINE CYLINDERS AS SHOWN BY TESTS OF THE NACA CYLINDER

By OSCAR W. SCHEY, VERN G. ROLLIN, and HERMAN H. ELLERBROCK, Jr.

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AERONAUTIC SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Abbreviation	Unit	Abbreviation
Length_____	l	meter_____	m	foot (or mile)_____	ft (or mi)
Time_____	t	second_____	s	second (or hour)_____	sec (or hr)
Force_____	F	weight of 1 kilogram_____	kg	weight of 1 pound_____	lb
Power_____	P	horsepower (metric)_____	_____	horsepower_____	hp
Speed_____	V	{kilometers per hour_____	kph	miles per hour_____	mph
		{meters per second_____	mps	feet per second_____	fps

2. GENERAL SYMBOLS

W Weight=mg

g Standard acceleration of gravity=9.80665 m/s² or 32.1740 ft/sec²

m Mass=$\dfrac{W}{g}$

I Moment of inertia=mk^2. (Indicate axis of radius of gyration k by proper subscript.)

μ Coefficient of viscosity

ν Kinematic viscosity

ρ Density (mass per unit volume)

Standard density of dry air, 0.12497 kg-m⁻⁴-s² at 15° C and 760 mm; or 0.002378 lb-ft⁻⁴ sec²

Specific weight of "standard" air, 1.2255 kg/m³ or 0.07651 lb/cu ft

3. AERODYNAMIC SYMBOLS

S Area

S_w Area of wing

G Gap

b Span

c Chord

A Aspect ratio, $\dfrac{b^2}{S}$

V True air speed

q Dynamic pressure, $\dfrac{1}{2}\rho V^2$

L Lift, absolute coefficient $C_L=\dfrac{L}{qS}$

D Drag, absolute coefficient $C_D=\dfrac{D}{qS}$

D_0 Profile drag, absolute coefficient $C_{D0}=\dfrac{D_0}{qS}$

D_i Induced drag, absolute coefficient $C_{Di}=\dfrac{D_i}{qS}$

D_p Parasite drag, absolute coefficient $C_{Dp}=\dfrac{D_p}{qS}$

C Cross-wind force, absolute coefficient $C_C=\dfrac{C}{qS}$

i_w Angle of setting of wings (relative to thrust line)

i_t Angle of stabilizer setting (relative to thrust line)

Q Resultant moment

Ω Resultant angular velocity

R Reynolds number, $\rho\dfrac{Vl}{\mu}$ where l is a linear dimension (e.g., for an airfoil of 1.0 ft chord, 100 mph, standard pressure at 15° C, the corresponding Reynolds number is 935,400; or for an airfoil of 1.0 m chord, 100 mps, the corresponding Reynolds number is 6,865,000)

α Angle of attack

ϵ Angle of downwash

α_0 Angle of attack, infinite aspect ratio

α_i Angle of attack, induced

α_a Angle of attack, absolute (measured from zero-lift position)

γ Flight-path angle

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By Oscar W. Schey, Vern G. Rollin, and Herman H. Ellerbrock, Jr.

SUMMARY

A method of constructing fins of nearly optimum proportions has been developed by the NACA to the point where a cylinder has been manufactured and tested. Data were obtained on cylinder temperature for a wide range of inlet-manifold pressures, engine speeds, and cooling-pressure differences.

The results indicate that an improvement of 40 percent in the outside-wall heat-transfer coefficient could be realized on the present NACA cylinder by providing a thermal bond equivalent to that of an integral fin-cylinder wall combination between the preformed fins and the cast cylinder wall.

Estimates over a range of wall thicknesses from 0.75 inch to 1.5 inches show that the pressure drop required for cooling could be decreased as much as 80 percent for a range of powers from present cruising to take-off power if the head fins on a present-day cylinder were replaced by fins of NACA cylinder proportions and if the thermal bond between the wall and the fins were equivalent to that of an integral arrangement. Calculations based on an inside-wall head temperature of 500° F indicate that the pressure drop required for cooling hypothetical cylinders of varying head-wall thickness could be decreased 50 percent by increasing the thickness from 0.7 inch to 1.5 inches. An estimate based on an inside-wall head temperature of 500° F and an integral fin bond indicates that the improved cooling obtained through the use of fins of NACA cylinder fin proportions will permit an increase of 35 percent in power output. This estimate of power increase is made solely for cooling criteria and does not include other factors limiting power output.

INTRODUCTION

The increase in power output of aircraft engines has been an important factor in the constantly improved performance of aircraft. The power output of air-cooled engines is limited in part by the quantity of waste heat that can be efficiently dissipated from the cylinder to the cooling-air stream. Improved cooling can be obtained either by increasing the quantity of cooling air by means of a blower or by increasing the cooling area by means of an improvement in fin proportions. An increase in the cooling area is the more efficient method from power and drag considerations. An analysis using equations obtained from heat-transfer tests (reference 1) showed that, if the heat-transfer coefficient for a constant outside-wall temperature is doubled by the choice of a suitable fin design, the resultant cooling capacity of the engine will be sufficient for a tripled power output.

Extensive investigations in which electrically heated test specimens were used have been conducted by the NACA to determine the fin proportions necessary for maximum heat transfer (reference 2). These investigations showed that the fins of service-type aircraft engines were not of optimum proportions. If the information obtained from this analysis were to be applied to the design of fins for aircraft cylinders, the power output would then probably be limited by factors other than cylinder cooling; that is, the design of spark plugs, the presence of preignition, the antiknocking quality of the fuel, or mechanical considerations. The results of these investigations were applied to the design of finning for an air-cooled engine cylinder. Because fins of the dimensions required to give a maximum heat transfer for a given width of finning and a given pressure drop across the baffles could not be produced by conventional machining and casting methods, a method of bonding preformed sheet-aluminum fins to the cast aluminum alloy of the cylinder head was developed by the NACA. Three progressively improved cylinders were constructed during 1937 and 1938 with approximately the same inside dimensions as a commercial cylinder in general use at that time. The fin proportions of this commercial cylinder were inadequate for cooling an engine with an increased power output.

The machining and casting methods for making fins have improved since the construction and testing of the NACA cylinders. Cylinders having a fin-surface area much greater than that of the commercial cylinder of 1937 are now in general use. No cylinder heads having a cooling area as large as that of the NACA cylinder head are known to have been cast or machined.

The present report describes the construction of the NACA cylinders and presents the results of the engine tests made with one of the cylinders. Equations are given for the average head and the average barrel temperatures of the NACA cylinder as functions of the fundamental engine and cooling variables. By means of these equations and similar ones for a commercial aircraft cylinder in general use in 1937 (hereinafter designated the A cylinder) and of approximately the same inside dimensions as the NACA cylinder, a comparison was made of the cooling performance of the two cylinders. The NACA cylinder was also compared with a commercial cylinder in general use in 1942 and of approximately the same inside dimensions as the A cylinder but with better finning (hereinafter denoted the B cylinder). Cooling data on the A cylinder were obtained from reference 1 and those

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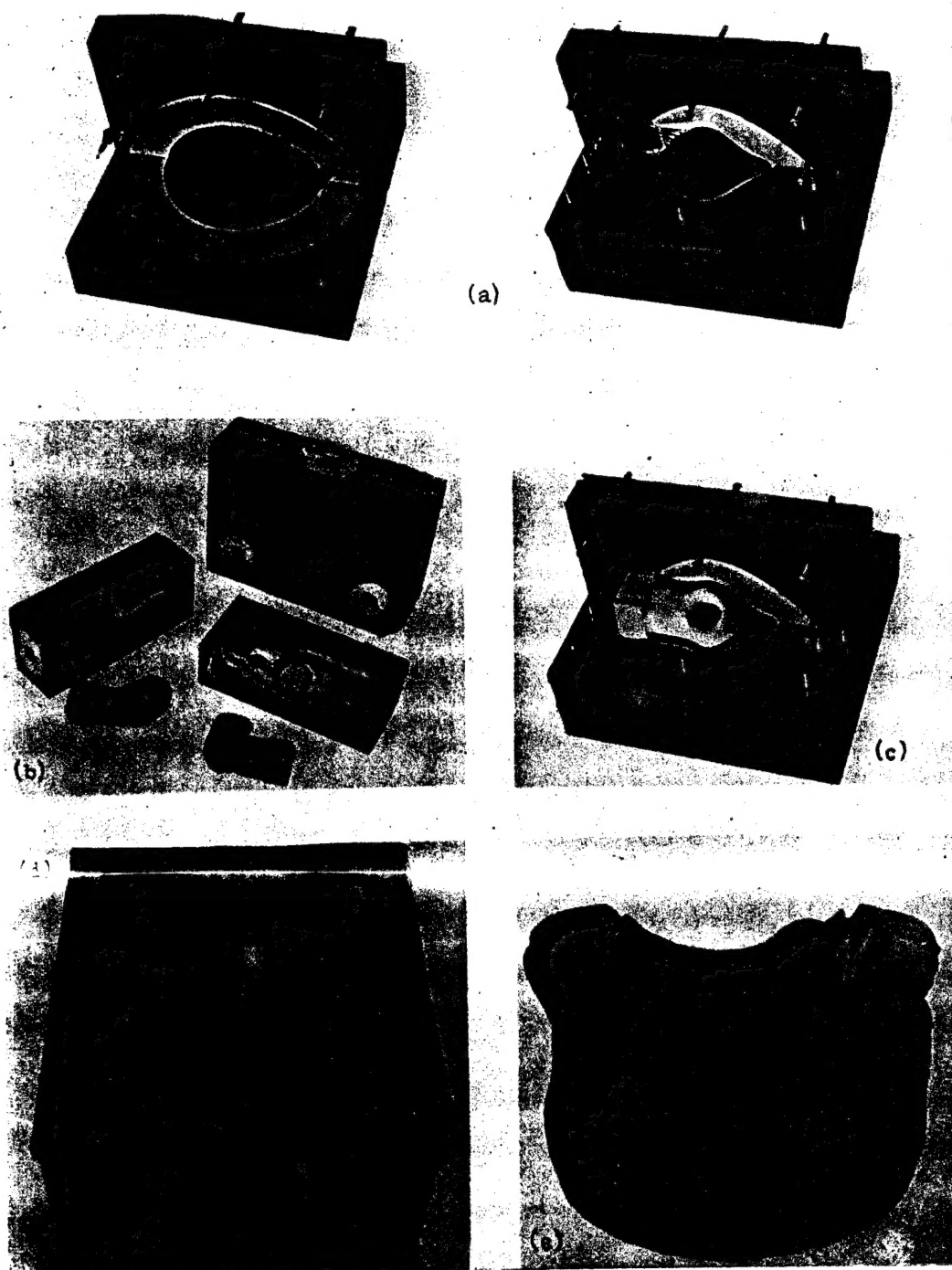
(a) Fins and spacers partly assembled.
(b) Cores constructed.   (c) Rocker-box inserted.
(d) Fins, spacers, and cores assembled in flask.   (e) Cylinder-head casting.

FIGURE 1.—Steps in the casting of the NACA cylinder head.

on the B cylinder, from unpublished tests made at Langley Memorial Aeronautical Laboratory in 1941.

Estimates were made of the decrease in pressure drop for a given power output and of the increase in power for a given pressure drop effected through the use of head fins of NACA cylinder proportions on the B cylinder. The fin bond was assumed to be thermally perfect, that is, to have the thermal properties of an integral fin-cylinder wall combination. This assumption was not realized in the construction of the NACA cylinder. All estimates of power increases are based solely on cooling considerations; mechanical limitation, fuel, or other factors might prevent the attainment of such powers.

Mr. Ernest Johnson, chief of the technical service division of LMAL, gave invaluable assistance in developing the technique for constructing the cylinders. Acknowledgment is made to Mr. Harry W. Lee, of the Norfolk Navy Yard, for his assistance in developing the technique of casting the NACA cylinders.

THE NACA CYLINDER

The results of an analysis of tests on electrically heated finned cylinder barrels enclosed in jackets showed that, for a large range of fin weights and for a pressure difference of 4 inches of water across the baffles, the optimum spacing for steel fins was approximately 0.07 inch and the optimum thickness, approximately 0.03 inch (reference 2). For aluminum fins the spacing was about the same but the thickness decreased to 0.02 inch. These proportions varied slightly with pressure difference. The analysis also indicated that the spacing and the thickness could be increased without an appreciable decrease in heat transfer. No similar tests have been made on cylinder heads but the results of the tests on the electrically heated cylinder barrels should be approximately applicable to cylinder heads.

After construction difficulties had been considered, a cylinder head having a fin spacing of 0.078 inch, a fin thickness of 0.031 inch, and a fin width of 1.9 inches was designed. At a pressure difference of 4 inches of water, these fin proportions would give 10 percent less heat transfer than fins of the same weight but of optimum proportions. In 1937 the commercial foundries considered it impracticable to construct the NACA cylinder head by casting the closely spaced wide fins. The fins were not machined on the head because the results were uncertain and the cost was prohibitive. The fins can be stamped from an aluminum alloy of high thermal conductivity and then attached to the wall. The use of such an alloy for the fins of a head cast by the conventional method is not possible because the fins must be of the same material as the wall, which requires an alloy of high strength with resulting reduced conductivity.

The preformed-fin method of construction was chosen and development work was started on small cylindrical specimens. The fins were stamped from an aluminum sheet and clamped together with interfin steel spacers leaving ¼ inch of the inner fin surface exposed to form the bond with the molten aluminum of the cylinder-head wall. The exposed fin surfaces were thoroughly cleansed to insure a good bond but, in spite of this precaution, an oxide film formed on the fins owing to the action of either the air or the molten aluminum on the fin. Specimens were made using various fluxes, various aluminum solders for tinning the exposed fin surface, and various materials for coating the surface of the spacers. The metal was poured with the mold spinning in an effort to scrub the oxide film off the fins. Both the preheat temperature of the mold and the pouring temperature of the metal were varied in order to determine the optimum temperatures. The results obtained with the specimens brought about the decision to cast the NACA cylinder head from an aluminum Y alloy with the mold stationary, with no flux, with a silica flour coating on the spacers, with the exposed fin surfaces tinned with an aluminum solder, and with a mold temperature of 1100° F and a pouring temperature of the aluminum of 1300° F.

Figure 1 shows the various steps in assembling the fins and the spacers and of preparing and assembling the cores. The thermocouples on the cylinder-head casting (fig. 1 (d)) were located on the exposed inner surface of the fins to indicate the mold temperature. The cylinder-head casting was heat-treated by heating to 950° F and quenching in water. The interfin spacers were left in place during the heat treatment to prevent the fins from warping.

The comparative finning of the A, the B, and the NACA cylinders is indicated in figure 2. The three cylinders have approximately the same inside dimensions, but the cylinder walls of the NACA cylinder head are much thicker than those of the other two cylinder heads. Cross sections of the A cylinder and the NACA cylinder are shown in figure 3. The NACA cylinder head is approximately 1½ inches thick at the heaviest section; the corresponding thickness of the A cylinder is 23/32 inch and of the B cylinder, 1 inch. The barrel of the NACA cylinder was constructed and the cylinder-head casting was machined by Pratt & Whitney Aircraft. The 39 steel barrel fins, 1/32 inch thick, 23/32 inch wide, and spaced 0.05 inch apart, were copper-brazed to the barrel wall. The barrel of the A cylinder has steel fins and the barrel of the B cylinder has an aluminum muff with aluminum fins.

The following table presents the fin areas, the fin weights, and the cylinder weights of the commercial and the NACA cylinders:

Cylinder	Fin-surface area (sq in.)		Weight of fins (lb)		Total weight (lb)	
	Head	Barrel	Head	Barrel	Barrel	Cylinder
A	1025	594	3.0	2.0	7.5	32.25
B	1575	1375	5.5	2.5	10.80	42.10
NACA	6727	1231	9.4	3.5	13.62	53.75

The fin-surface area on the head of the NACA cylinder is considerably larger than that on the head of either of the commercial cylinders. Although the NACA cylinder has about four times as much head fin-surface area as the B cylinder, the difference in weight due to these fins is only about 4 pounds. The present fin bond of the NACA cylinder is ¼ inch deep and adds 5¼ pounds of weight; a ⅛-inch bond could be achieved, thereby further reducing the weight of the cylinder.

A   B   NACA

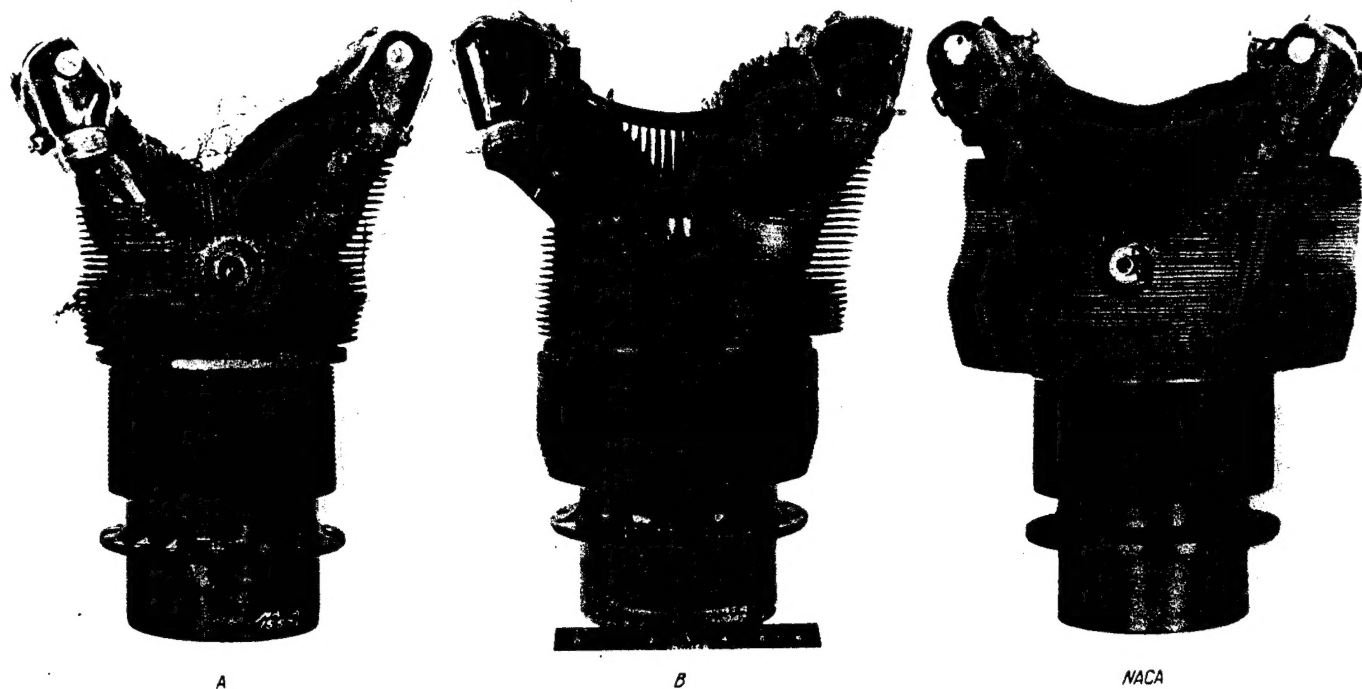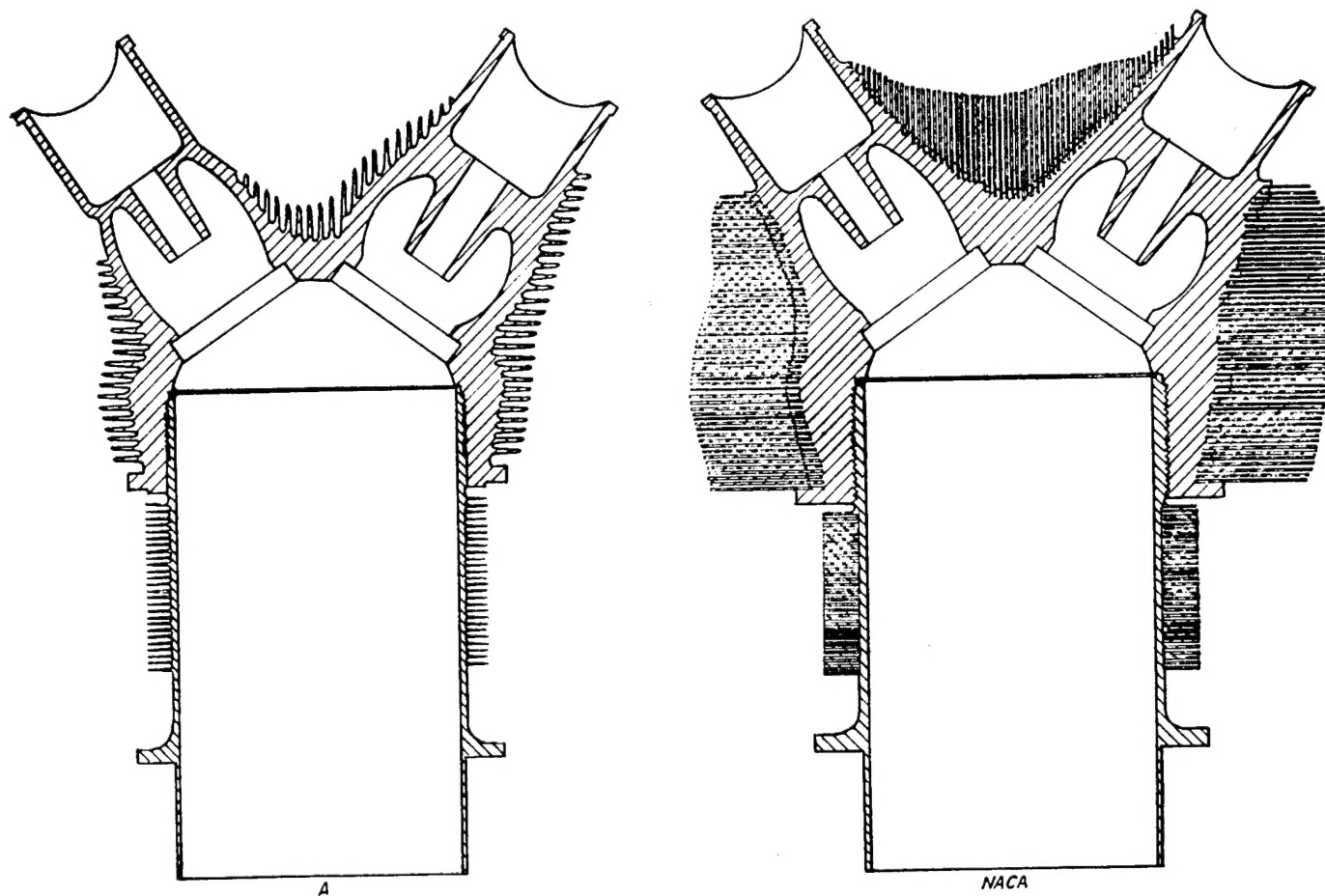FIGURE 2.—A and B commercial cylinders and NACA cylinder.

A   NACA

FIGURE 3.—Cross sections of A and NACA cylinders.

APPARATUS

TEST SETUP

A diagrammatic sketch of the single-cylinder test unit is shown in figure 4 and a photograph, in figure 5. The engine has a bore of 5¾ inches, a stroke of 6 inches, and a compression ratio of 5.9. The valve timing, approximately that of the standard commercial engines, is:

Intake opens, degrees B. T. C. _____ 20
Intake closes, degrees A. B. C. _____ 70
Exhaust opens, degrees B. B. C. _____ 70
Exhaust closes, degrees A. T. C. _____ 20

In all the tests a single-cylinder injection pump and an injection valve located in a hole in the cylinder above the front spark plug were used. Six equally spaced holes of 0.063-inch diameter were drilled in the cylinder liner below the cylinder flange for the introduction of lubricating oil under 2 pounds pressure. This additional lubrication was needed because of the long cylinder adapter and connecting rod. The cooling system consisted of an orifice tank, a centrifugal blower, an air duct, and a sheet-metal jacket enclosing the cylinder. A diagram of the jacket is shown in figure 6. The jacket had a wide entrance section, which gave a low air velocity in front of the cylinder; the rear half of the jacket fitted closely against the fins, which permitted effective use of the air. The jacket-exit areas for both the head and the barrel were 1.6 times the free-flow area between the fins. A partition placed in the rear of the jacket separated the air that flowed over the head from the air that flowed over the barrel.

An auxiliary blower was used to provide inlet-manifold pressures above atmospheric pressure. A surge tank was installed in the combustion-air system above the engine to reduce pulsations. The engine power was absorbed by a water brake interconnected with an electric dynamometer; the torque was read from dial scales. The NACA standard test-engine equipment was used to determine engine speed and fuel consumption.

INSTRUMENTS

Iron-constantan thermocouples and a potentiometer were used to measure the cylinder temperatures. The thermocouples of enameled and silk-covered 0.016-inch-diameter wire were peened into the outer surface of the cylinder head and spot-welded to the outer surface of the barrel. The lo-

A, orifice
B, surge tank
C, water manometer
D, water manometer
E, static-pressure manometer
F, thermometer
G, thermocouples

H, line from blower
I, two-way valve
J, injection pump
K, phase-changing gear
L, thermometer
M, thermometer
N, surge tank

O, cowling
P, fuel-air-ratio indicator
Q, exhaust-gas sample
R, inlet-pressure manometer
S, orifice differential
T, orifice total pressure
U, water brake

V, thermometer
W, selector switch
X, dynamometer
Y, engine
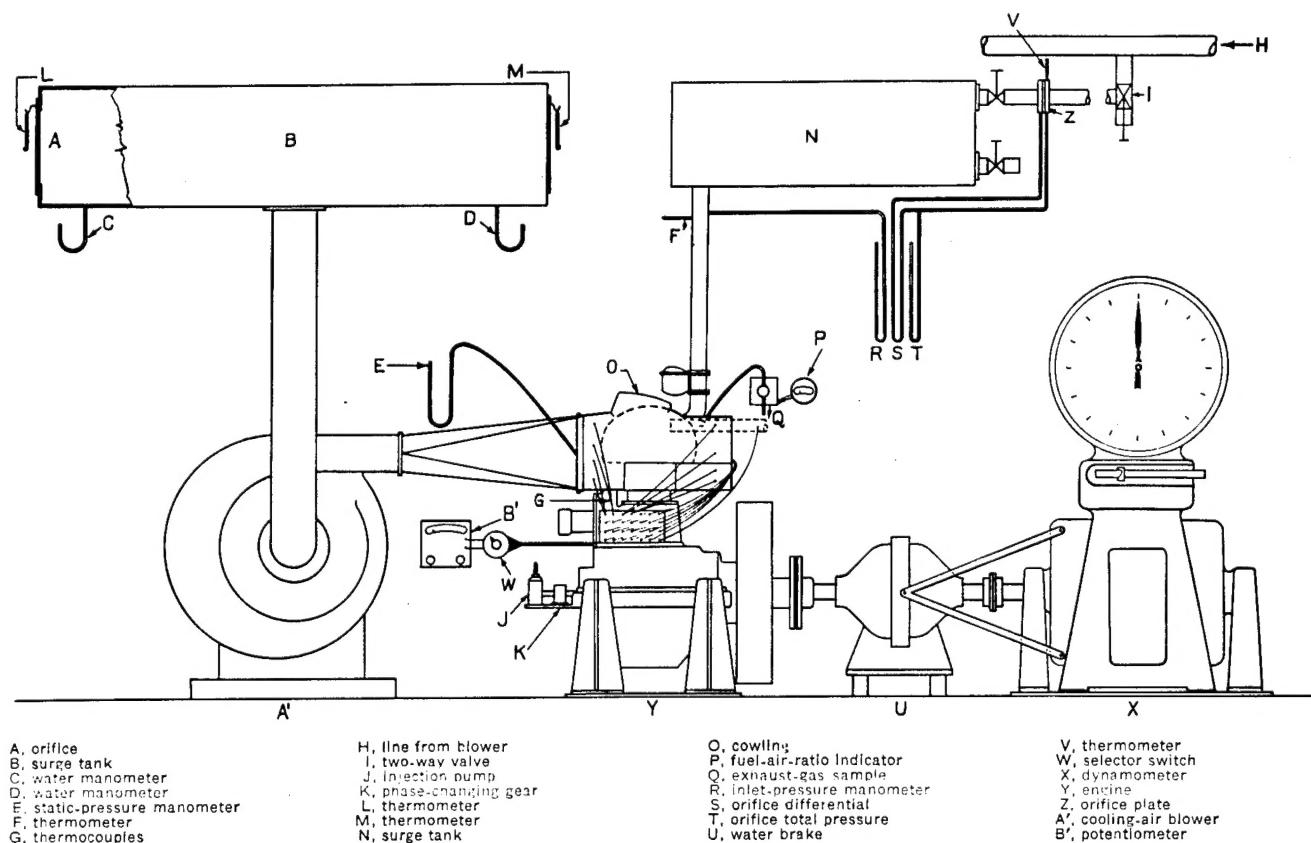Z, orifice plate
A', cooling-air blower
B', potentiometer

FIGURE 4.—Diagrammatic layout of equipment.

cations of the 22 thermocouples on the cylinder head, the 10 on the barrel, and the 2 on the flanges are shown in figure 7. In addition, 10 thermocouples, differentiated by the suffix A, were located in several fins as close as possible to the cylinder wall and adjacent to the similarly numbered thermocouples to measure the temperature drop from the wall to the fin.

The temperatures of the cooling air passing over the head and the barrel were separately measured. The cooling-air temperatures were measured at the jacket entrance near the cylinder by two multiple thermocouples consisting of two thermocouples electrically connected in series, and at the outlet of the jacket by two multiple thermocouples consisting of four thermocouples electrically connected in series. The cold junctions of all thermocouples were placed in an insulated box. Liquid-in-glass thermometers were used to measure the cold-junction temperature, the air temperatures at the Durley thin-plate orifices, and the air temperatures in the inlet manifold near the cylinder.

The pressure drop across the cylinder was measured by a static ring around the air duct located ahead of the engine cylinder where the velocity head was negligible (fig. 6). The static ring was connected to a water manometer. An inclined water manometer was used to measure the orifice-tank pressure and a mercury manometer, to measure the inlet-manifold pressure.

The weight of combustion air was determined with a thin-plate orifice and a multiple manometer that indicated the total pressure in inches of mercury and the pressure drop across the orifice in inches of water. The fuel consumption was determined from measurements of the time required to consume a fixed weight of fuel. Accuracy was obtained by using a sensitive balance that electrically operated a stop watch. A Cambridge fuel-air-ratio meter was used for the convenience of the operator, but the actual mixture ratios were calculated from measurements of air and fuel consumption.

TESTS AND COMPUTATIONS

Tests of the NACA cylinder were conducted to determine the constants used in equations for the average head and the average barrel temperatures as functions of the funda-
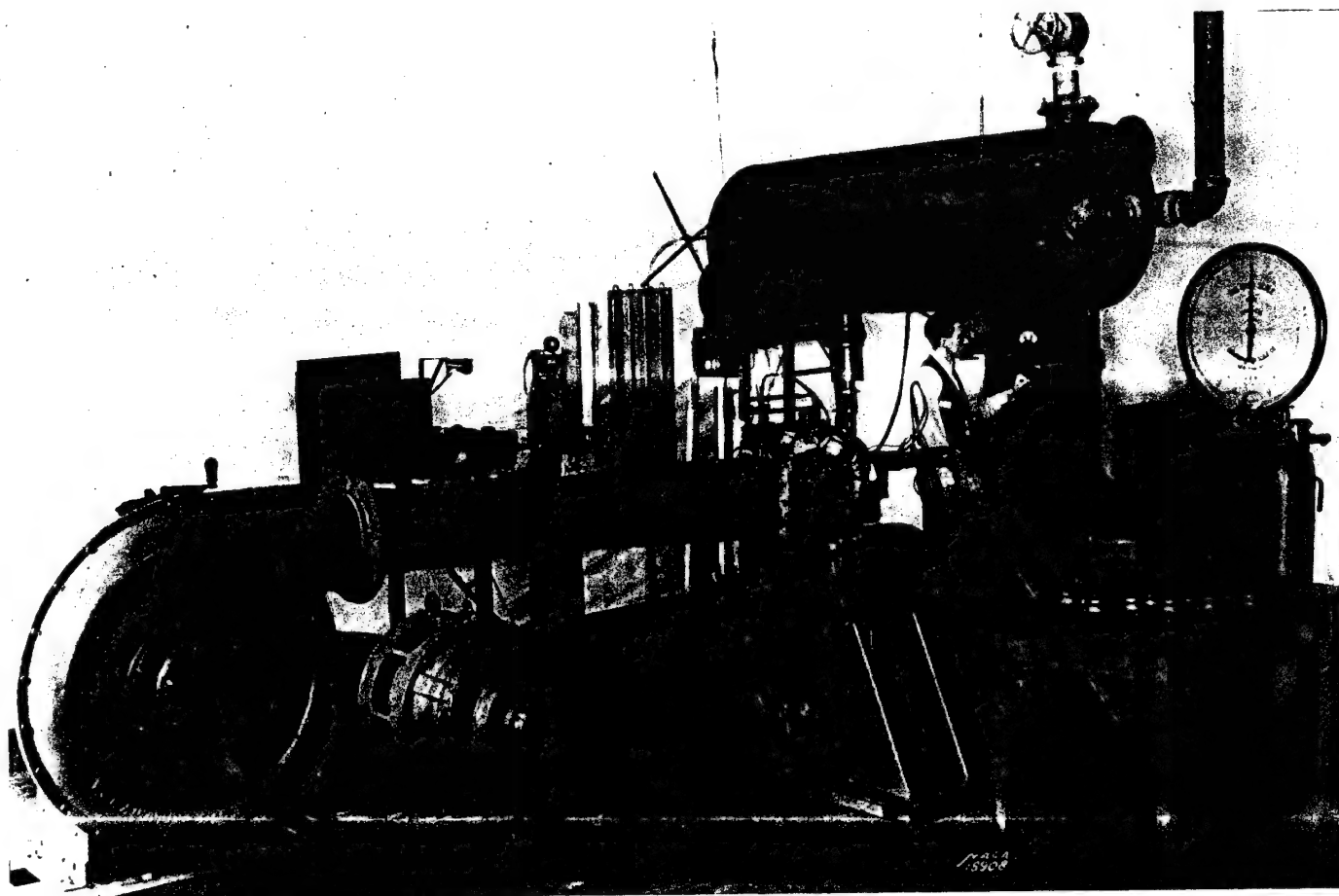

FIGURE 5.—Setup of single-cylinder air-cooled engine.

mental engine and cooling variables. In order to obtain the constants in the equations for the average head and barrel

Section A–A

FIGURE 6.—Diagram of wide-entrance jacket enclosing NACA cylinder.

temperatures, the data were calculated and plotted by the method of reference 1. The constants in the equations used to determine heat-transfer coefficients from the gas to the inside wall were determined from tests in which a jacket slightly different from that shown in figure 6 was used. Inside-wall coefficients, however, are independent of jacket form and the coefficients determined in the foregoing tests are applicable to the wide-entrance jacket.

From the tests of the NACA cylinder and the data from similar tests that have been made on the A and the B cylinders, cylinder temperatures, power required for cooling, pressure drop required for cooling, heat-transfer coefficients, and heat dissipated to the cooling air were computed for the three cylinders. The shape of the jackets used on the A and the B cylinders was the same as that used on the NACA cylinder. More tests were made on the NACA cylinder than were necessary to establish the values of the constants in the equations; the additional tests were made to check the validity of the values over a range of engine and cooling conditions.

Calibration tests were made to determine the weight of air flowing over the head and the barrel of the NACA cylinder as a function of the pressure drop across the cylinder (fig. 8). The pressure drop Δp obtained from the static ring placed in front of the cylinder included both the drop across the cylinder and the loss at the jacket exit. The pressure drop is given in inches of water and is corrected to a standard air density of 70° F and 29.92 inches of mercury absolute.
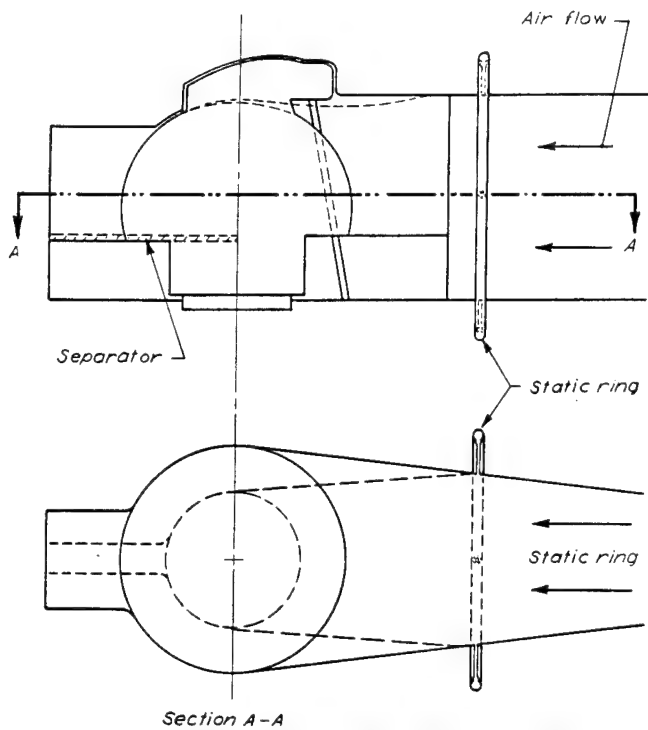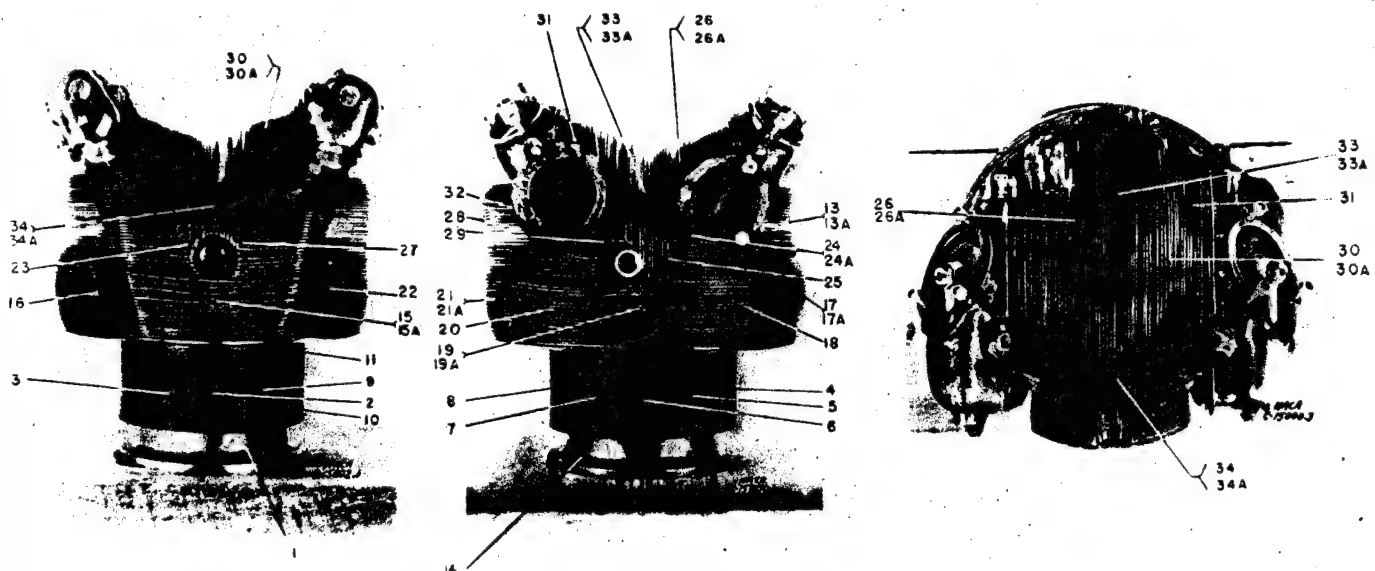

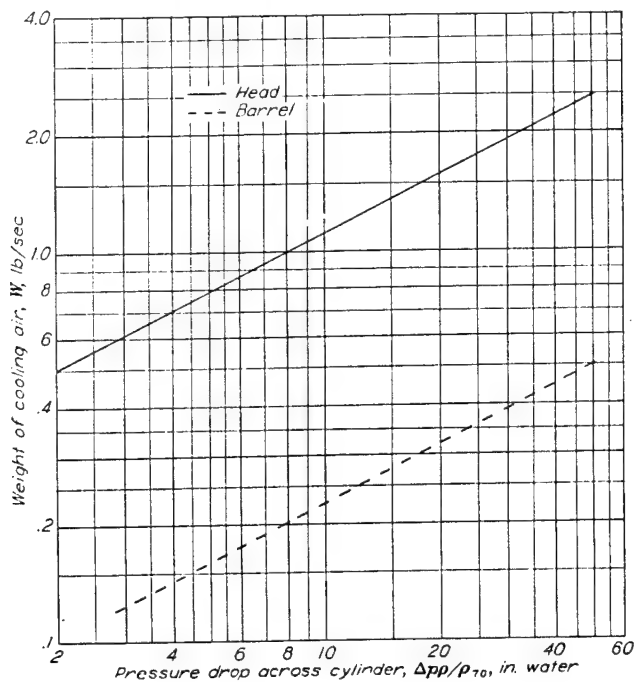FIGURE 7.—Three views of NACA cylinder showing location of thermocouples.

FIGURE 8.—Variation of cooling-air weight with pressure drop across head and barrel of cylinder.

The method of computing the cooling-air weight from the pressure drop across the Durley orifices is given in reference 3. The engine brake horsepower was calculated from corrected dynamometer-scale readings and the engine speed. The friction horsepower was determined by motoring the engine at the inlet-manifold pressures, the exhaust pressures, and the speeds used in the power runs. The indicated horsepower was obtained by adding the brake horsepower to the friction horsepower.

The tests covered the following range of conditions:

Cooling-air temperature, °F	89–108
Brake mean effective pressure, pounds per square inch	83–190
Engine speed, rpm	1500–2100
Pressure drop across cylinder, inches of water	2.4–33.3
Fuel-air ratio	0.079–0.080
Spark timing, degrees B. T. C.	26–31
Inlet-air temperature, °F	80–92

In all of the tests except those in which the brake mean effective pressure was varied, the weight of the combustion air delivered to the engine per cycle was held constant. The weight of combustion air was calculated from the pressure and the temperature readings at the thin-plate orifice inserted in the duct between the blower and the engine, using orifice coefficients and equations from reference 4. The test data obtained with the NACA cylinder are plotted in figures 9, 10, 11, and 18.

Analysis of a large number of tests for several cylinders (references 1 and 5) led to a choice for the effective gas temperature T_g, which enters in the equations for average head and barrel temperatures, of 1150° F for the head and 600° F for the barrel. These values are for inlet-air temperatures

of approximately 80° F, fuel-air ratios near the chemically correct mixture, and normal spark settings. These values of T_g were used in the present report to determine the constants in the cooling equations of the NACA cylinder for average head and barrel temperatures as functions of the engine and the cooling variables. For estimates of pressure drop required for cooling and of power obtainable from the A, the B, and the NACA cylinders for mixtures other than the chemically correct mixture, values of T_g were obtained from tests conducted at Langley Memorial Aeronautical Laboratory on the B cylinder.

Gasoline conforming to Army specification No. 2–92, grade 100 (100-octane number, Army method) was used for all the tests.

RESULTS AND DISCUSSION
COOLING EQUATIONS

Constants for the fundamental cooling equations are obtained from the curves in figures 9 and 10 by methods described in reference 1. The curves for the cylinder head in figures 9 and 10 are based on the temperatures measured by the thermocouples on the outside-wall surface of the head. The data for figure 9 were obtained from tests using a different jacket, as previously mentioned, but the constants in the equations obtained from these curves are applicable to the cooling equation for the cylinder with the wide-entrance jacket of figure 6. The equations for the NACA cylinder with the wide-entrance jacket are:

$$T_h - T_a = \frac{T_g - T_a}{\dfrac{0.670 a_0 (\Delta p \rho / \rho_{70})^{0.2}}{0.0356 a_1 I^{0.72}} + 1} \qquad (1)$$

$$T_b - T_a = \frac{T_g - T_a}{\dfrac{0.647 a_0 (\Delta p \rho / \rho_{70})^{0.33}}{0.0159 a_1 I^{0.72}} + 1} \qquad (2)$$

where

T_h average temperature over outside cylinder-head surface when equilibrium is attained, °F

T_b average temperature over outside cylinder-barrel surface when equilibrium is attained, °F

T_a inlet temperature of cooling air, °F

T_g effective gas temperature, °F

a_0 outside-wall area of head (or barrel) of cylinder, square inches
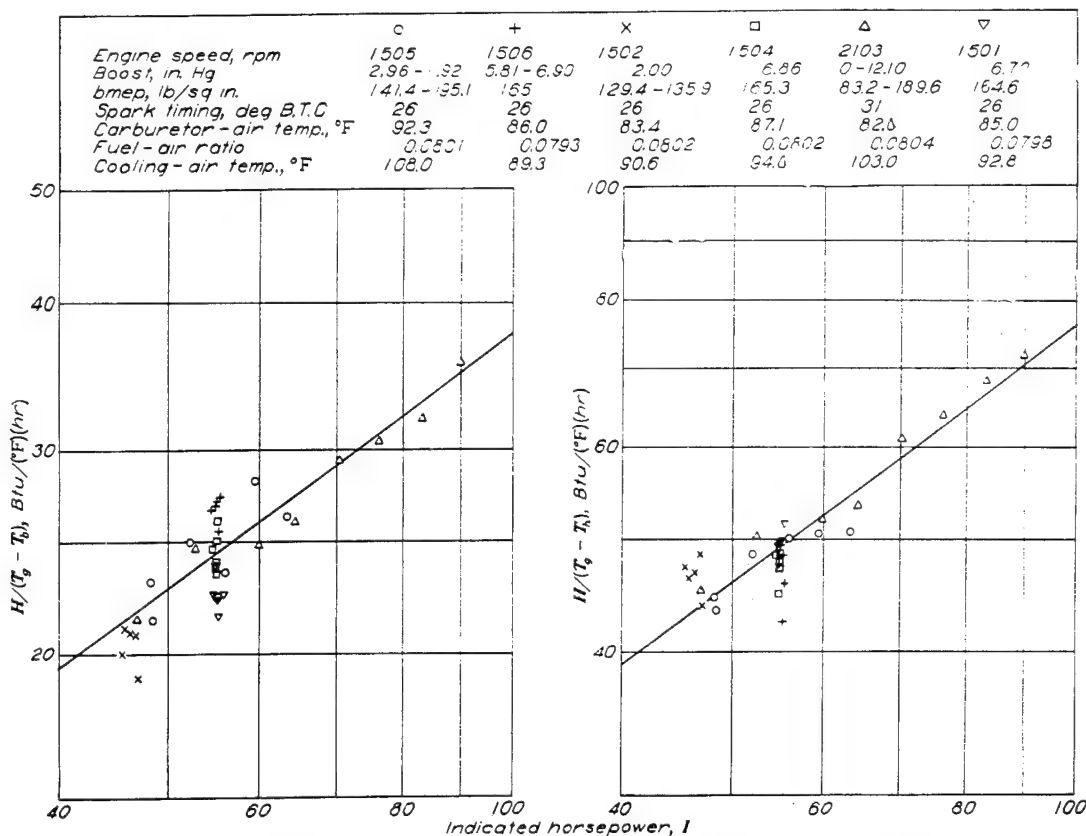
a_1 internal area of head (or barrel) of cylinder, square inches

Δp pressure drop across cylinder including loss from jacket exit, inches of water

ρ average density of cooling air entering and leaving fins, pound feet^{-4} second ^{2}

ρ_{70} density of air at 29.92 inches of mercury and 70° F, pound feet^{-4} second ^{2}

I indicated horsepower per cylinder

	○	+	×	□	△	▽
Engine speed, rpm	1505	1506	1502	1504	2103	1501
Boost, in. Hg	2.96 - .92	5.81 - 6.90	2.00	6.86	0 - 12.10	6.72
bmep, lb/sq in.	141.4 - 95.1	155	129.4 - 135.9	165.3	83.2 - 189.6	164.6
Spark timing, deg B.T.C.	26	26	26	26	31	26
Carburetor - air temp., °F	92.3	86.0	83.4	87.1	82.8	85.0
Fuel - air ratio	0.0801	0.0793	0.0802	0.0802	0.0804	0.0798
Cooling - air temp., °F	108.0	89.3	90.6	94.6	103.0	92.8

FIGURE 9.—Variation of $H/(T_g-T_A)$ and $H/(T_g-T_b)$ with indicated horsepower for NACA cylinder.
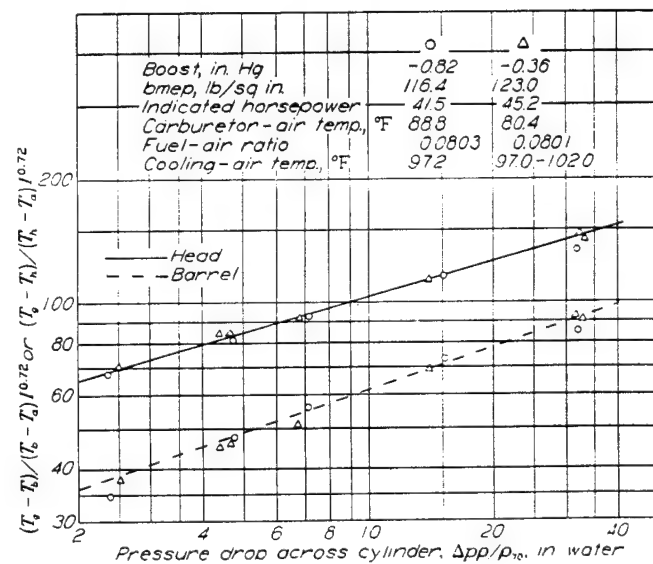
A temperature of 70° F instead of the usual 60° F was used in calculating standard density to facilitate comparisons with earlier data on cylinder cooling that had been based on a temperature of 70° F.

The areas of the A, the B, and the NACA cylinders are listed in table I.

TABLE I.—AREAS OF THREE CYLINDERS

Cylinder	Internal-wall area, a_i (sq in.)		Exterior-wall area, a_0 (sq in.)	
	Head	Barrel	Head	Barrel
A	76.8	84.8	142	68.5
B	78.4	85.5	145	72.2
NACA	76.8	84.8	218	62.6

The methods of obtaining a_0 and a_i are given in reference 1. All three cylinders show greater internal-wall areas than exterior-wall areas on the barrels, which may seem incorrect. The exterior-wall areas of the barrels are, however, based on a length that is slightly greater than the barrel length covered by the fins; whereas, the internal-wall areas are based on a length measured from the bottom of the head flange to the position of the bottom compression ring when the piston is at bottom center. The exterior-wall areas of the barrels are chosen for use in computations because this area is the one over which the cooling air flows. Heat is

	○	△
Boost, in. Hg	-0.82	-0.36
bmep, lb/sq in.	116.4	123.0
Indicated horsepower	41.5	45.2
Carburetor - air temp., °F	88.8	80.4
Fuel - air ratio	0.0803	0.0801
Cooling - air temp., °F	97.2	97.0,-102.0

FIGURE 10.—Variation of $\left(\frac{T_g-T_b}{T_b-T_a}\right) I^{0.72}$ and $\left(\frac{T_g-T_h}{T_h-T_a}\right) I^{0.72}$ with pressure drop across NACA cylinder. Engine speed, 1501 rpm; spark timing, 26° B. T. C.

carried away by free convection from the small area between the flange and the cylinder jacket. This heat quantity is too negligible to be considered in the cooling equations.

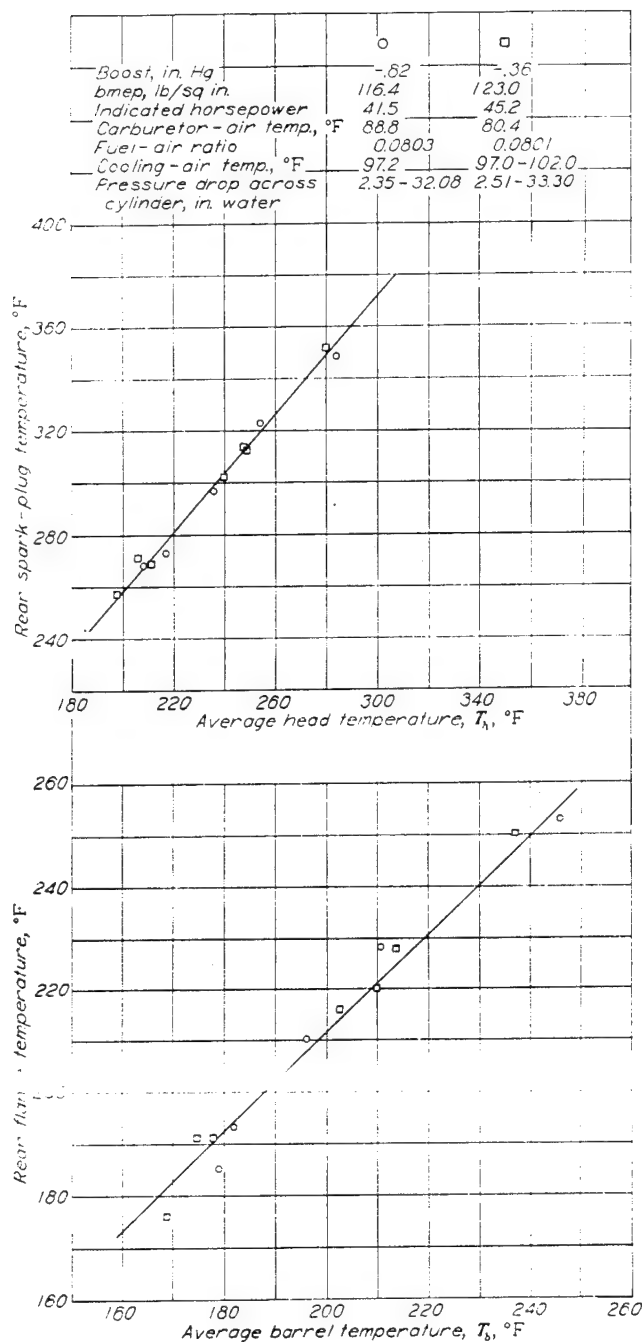|	○	□
Boost, in. Hg	-.82	-.36
bmep, lb/sq in.	116.4	123.0
Indicated horsepower	41.5	45.2
Carburetor-air temp., °F	88.8	80.4
Fuel-air ratio	0.0803	0.0801
Cooling-air temp., °F	97.2	97.0-102.0
Pressure drop across cylinder, in. water	2.35-32.08	2.51-33.30

FIGURE 11.—Variation of rear spark-plug temperature with average head temperature, and rear flange temperature with average barrel temperature for NACA cylinder. Engine speed, 1501 rpm; spark timing, 27° B. T. C.

CYLINDER-TEMPERATURE RELATIONSHIPS

In reference 6 equations are developed in which temperatures at individual points on the cylinder are taken as functions of the engine and the cooling conditions. To determine temperatures at individual points by such equations is tedious; if some simple relation between the average cylinder temperatures and pertinent individual temperatures were obtained, only the equations for the average temperatures and these simple relations must be known to determine the individual temperatures.

The rear spark-plug temperature and the rear flange temperature are the most pertinent temperatures of a cylinder because they are the present criterion of cylinder cooling for multicylinder engines. The rear spark-plug temperature and the rear flange temperature of the NACA cylinder have been plotted in figure 11 against the average head and the average barrel temperatures for the tests that were made with the wide-entrance jacket. Because the construction of the single-cylinder test jacket does not permit the passage of cooling air over the flange of the cylinder, the rear-flange, average-barrel temperature relationship is not representative of multicylinder results. A linear relationship exists for both sets of temperatures. Although there was very little variation of engine conditions for the tests shown in figure 11, the tests of a commercial cylinder previously mentioned have shown that, when this relationship is established for two temperatures, it will be applicable for any engine or cooling condition. Figure 11 will therefore be used to determine T_b and T_h when the rear spark-plug and the rear flange temperatures are known for the engine and cooling conditions.

For a given rear spark-plug temperature, an analysis of the cooling equations for the head and the barrel of the NACA cylinder and of the curve for average head temperature plotted against rear spark-plug temperature shows that, although T_h is constant for any engine and cooling condition, T_b does not remain constant as engine and cooling conditions are varied. The only case in which T_b remains constant, when the rear spark-plug temperature is constant but the cooling and the engine conditions vary, occurs when the values of exponents m and n' are the same in both the head and the barrel equations. From figure 11, the limiting value of T_h to be used in the cooling equations for determining estimated pressure drops or horsepowers can be obtained if the limiting value of the rear spark-plug temperature is known.

COMPARISON OF COOLING PERFORMANCE OF A, B, AND NACA CYLINDERS

Comparison of indexes of cooling. Equations (1) and (2) and similar equations for the A cylinder from reference 1 and for the B cylinder from unpublished data were used in obtaining the curves shown in figure 12 for values of $\Delta p\rho/\rho_{70}$ of 2 and 30 inches of water. The index of cooling is the value of the ratios $(T_h-T_a)/(T_g-T_h)$ and $(T_b-T_a)/(T_g-T_b)$. The lower the values for a given pressure drop across the cylinder, the better the cooling. Comparisons of the cooling of different cylinders of the same displacement have generally been made at LMAL by plotting the index of cooling against $I^2/(\Delta p\rho/\rho_{70})$; one curve for the head and one for the barrel of a cylinder have been applicable to all pressure drops. The nature of the equation, in which K_1 is a constant,

$$\frac{T_h-T_a}{T_g-T_h}=\frac{1}{K_1}\left[\left(\frac{I^2}{\Delta p\rho/\rho_{70}}\right)^{\frac{n'}{2}}(\Delta p\rho/\rho_{70})^{\frac{n'}{2}-m}\right]$$

shows that, if m equals $n'/2$, one curve of $\frac{T_h-T_a}{T_g-T_h}$ plotted against $\frac{I^2}{\Delta p\rho/\rho_{70}}$ will be applicable to all pressure drops. The difference between exponents $n'/2$ and m for the NACA cylinder is not negligible; for that reason separate curves are given in figure 12 for each pressure difference.

Figure 12 shows a marked improvement in cooling for the NACA cylinder as compared with the A cylinder for both the head and the barrel when the cooling is based on average outside-wall temperature. The inside-wall temperatures will be discussed later. The NACA cylinder head also shows improvement over the head of the B cylinder, especially at low pressure differences. That the NACA cylinder barrel shows slightly better cooling on the basis of operating temperatures than the B cylinder barrel was unexpected because the B cylinder barrel has 11 percent more fin area than the NACA cylinder barrel. Furthermore, the fins on the B cylinder barrel are of aluminum and therefore have a higher conductivity than the steel fins on the NACA cylinder barrel. The reason for these results is, as will be shown later in the report, that the B cylinder barrel dissipates more heat than the NACA cylinder barrel. It is therefore misleading to compare the two cylinder barrels on the basis of temperature and neglect the total amount of heat each dissipated. For the same reason, the NACA cylinder head would be slightly better from the standpoint of cooling than is shown in figure 12.

Comparison of cooling pressure drops.—A better representation of the relative cooling of the three cylinders than that shown in figure 12 is shown in figures 13 and 14. The criterion for satisfactory cooling has been assumed to be 450° F on the rear spark plug and the calculations have been based on cooling air at a pressure of 29.92 inches of mercury absolute and a temperature of 100° F, which are the values for Army standard air at sea level. Figure 13 shows the pressure drop required for cooling at various indicated horsepowers. The values of horsepower cover a range from approximately rated power of the A cylinder to the approximate power developed in take-off at the present time. The curves for the NACA cylinder were obtained by finding the average head temperature from figure 11 and the rear spark-plug temperature and then determining the pressure drop from equation (1). The density of the inlet cooling air was used instead of the average density in equation (1) because use of the density from equation (1) would have complicated
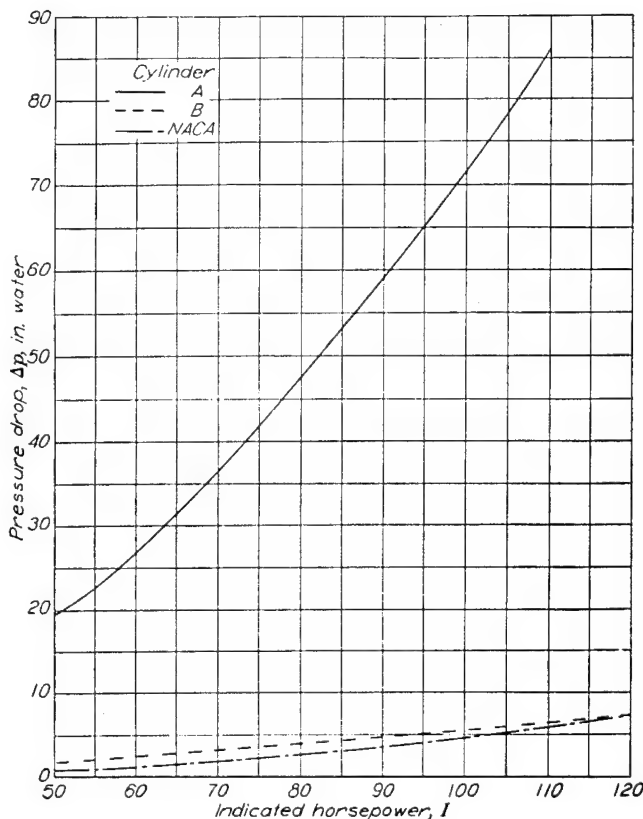

FIGURE 13.—Pressure drop required to cool A, B, and NACA cylinders for rear spark-plug temperature of 450° F. Cooling-air pressure at inlet, 29.92 inches Hg absolute; cooling-air temperature, 100° F; fuel-air ratio, 0.08.

the calculations and would have changed the results only slightly. The curves for the other two cylinders were found by a similar method.

From an extrapolation of figure 13, a pressure drop of about 100 inches of water would be required to cool the A cylinder with present take-off powers; whereas the B cylinder and the NACA cylinder will cool under similar conditions with a pressure drop of 7 inches of water. Figure 12 shows that the NACA cylinder head is much cooler than the B cylinder head for given engine and cooling conditions, yet figure 13 shows comparatively little improvement in pressure drop required by the NACA cylinder. Figure 12, however, is based on an average outside-wall temperature, whereas figure 13 is based on a constant rear spark-plug temperature. In order to obtain a rear spark-plug temperature of 450° F on the NACA cylinder, the average head temperature must be held at 368° F; the average head temperature of the B cylinder is 429° F for a rear spark-plug temperature of 450° F. Thus, in figure 13 the head of the NACA cylinder is much cooler than the head of the B cylinder. Indexes of cooling other than rear spark-plug temperature must therefore be established in order to compare cylinders of widely different geometric design. If $\bar{B}$, n', k_m, t_w, T_h, T_g, and the cylinder surface areas are known, the inside-wall temperature of the head of a cylinder can be calculated for various horsepowers by the equation developed in the appendix.
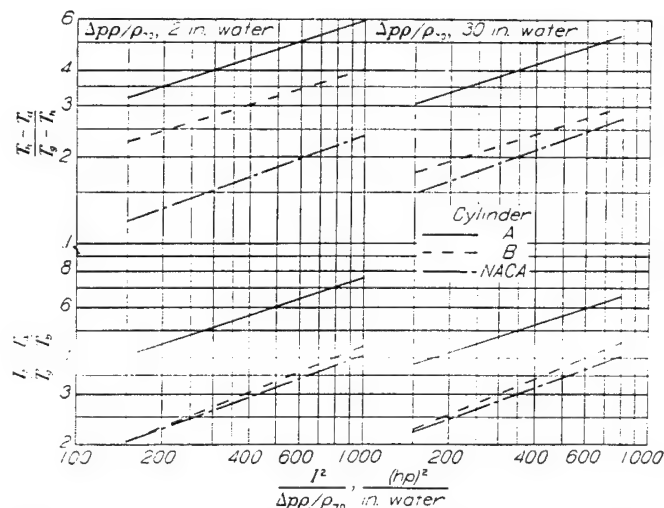

FIGURE 12.—Comparative cooling of A, B, and NACA cylinders for 2 and 30 inches of water pressure drop across cylinder. Head T_g, 1150° F; barrel T_g, 600° F.

$$T_{h,1} = \frac{\bar{B} a_1 I^{n'} (T_g - T_h)}{a_1 k_m / t_w} + T_h \qquad (6)$$

where

$\overline{B}$ and n' constants
k_m thermal conductivity
t_w average wall thickness

The inside-wall temperatures of the NACA, the A, and the B cylinders in figure 14 were calculated for the same conditions for the pressure-drop calculations of figure 13. As the horsepower increases, the inside-wall temperature of the three cylinders increases for a constant rear spark-plug temperature of 450° F. For low powers, the inside-wall temperatures of the NACA cylinder are lower than those of the B cylinder, but for high powers the inside-wall temperatures of the NACA cylinder are higher than those of the B cylinder. The NACA cylinder head is about 1.5 inches thick and the B cylinder head is about 1 inch thick. The greater area of the NACA cylinder head facilitates heat transfer but the increased thickness counteracts this effect. The two factors combine to give the results shown in figure 14. The barrel temperatures of the three cylinders shown in figure 14 were obtained by substituting the horsepowers
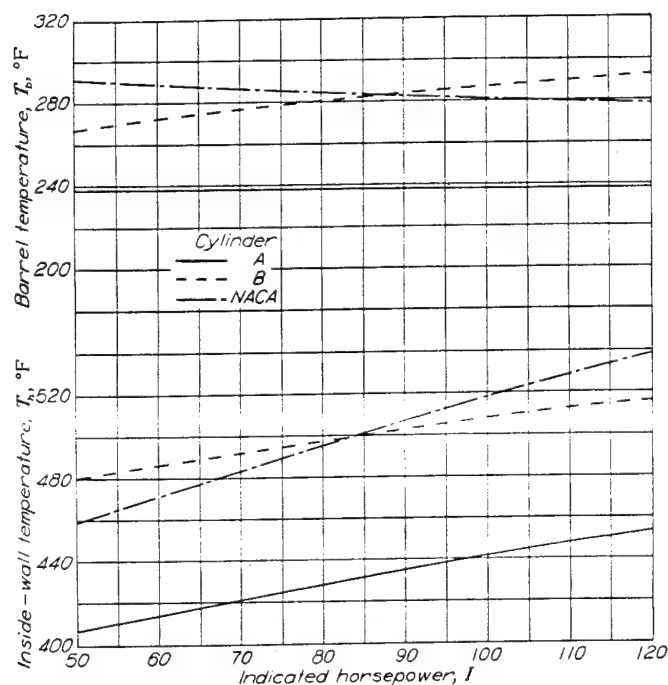

FIGURE 14.—Barrel and inside-wall temperatures of A, B, and NACA cylinders for rear spark-plug temperature of 450° F. Cooling-air temperature, 100° F; cooling-air pressure, 29.92 inches Hg absolute; fuel-air ratio, 0.08.

and the pressure drops of figure 13 in equation (2) and in similar equations for the A and B cylinders.

Because both the inside-wall and the outside-wall temperatures of the three cylinders were different when the index of cooling was 450° F on the rear spark plug, and because preignition, knock, and cylinder strength all depend on inside-wall temperature, an estimate of the pressure drop required for cooling the cylinder heads was made on the basis of a constant inside-wall temperature of 500° F.

The pressure drop required for cooling based on constant inside-wall temperature is given by

$$\Delta p = \frac{\rho_{70}}{\rho} \left\{ \frac{\overline{B} a_1 I^{n'} k_m (T_{h,1} - T_g)}{K a_0 [\overline{B} I^{n'} t_w (T_g - T_a) - k_m (T_{h,1} - T_a)]} \right\}^{1/m} \quad (7)$$

in which m is a constant. (See appendix for derivation.) Equation (7) was used to determine the curves of figure 15, which show that, when the criterion of cooling is an inside-wall head temperature of 500° F, less pressure drop is needed to cool the A cylinder than when the criterion of cooling is a rear spark-plug temperature of 450° F (fig. 13); the differential amounts to about 45 inches of water as compared with 100 inches of water. The constants in equation (7) were obtained from equation (1) for the NACA cylinder, from reference 1 for the A cylinder, and from unpublished data for the B cylinder. The calculations were based on Army standard sea-level pressure and temperature, and the density ρ in the equation was assumed to be inlet density. The temperature of the inside wall must have been less than 500° F for the A cylinder with 450° F on the rear spark plug. The B cylinder needs a slightly higher pressure drop to hold 500° F on the inside wall of the head
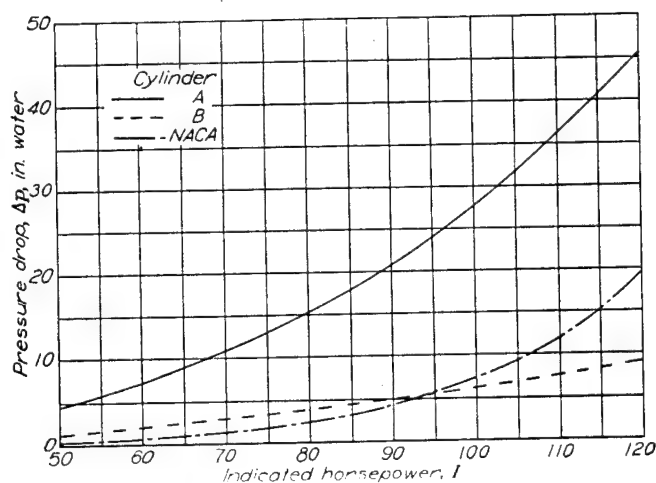

FIGURE 15.—Pressure drop required to cool A, B, and NACA cylinders for inside-wall temperature of 500° F. Cooling-air pressure at inlet, 29.92 inches Hg absolute; cooling-air temperature, 109° F; fuel-air ratio, 0.08.

than to hold 450° F on the rear spark plug. The increase, however, is not very great; a horsepower of 120 requires about 9 inches of water pressure drop to hold 500° F on the inside wall as against 7.4 inches of water to hold 450° F on the rear spark plug. The NACA cylinder pressure drops increased to about 20 inches of water in order to hold 500° F on the inside wall as compared with about 7 inches of water to hold 450° F on the rear spark plug. This large pressure drop in the case of the NACA cylinder is principally due to the poor thermal bond between the fins and the head. The increased wall thickness of the NACA cylinder head over that of the B cylinder tends to increase the required pressure drop, whereas the increased outside-wall area due to the increased thickness tends to decrease the required pressure drop. The separate effects of wall thickness and outside-wall area on cooling will be discussed later. The corresponding barrel temperatures of the three cylinders for the conditions given in figure 15 are shown in figure 16. The high pressure drops required to cool the heads of the NACA and the A cylinders make the barrel temperatures
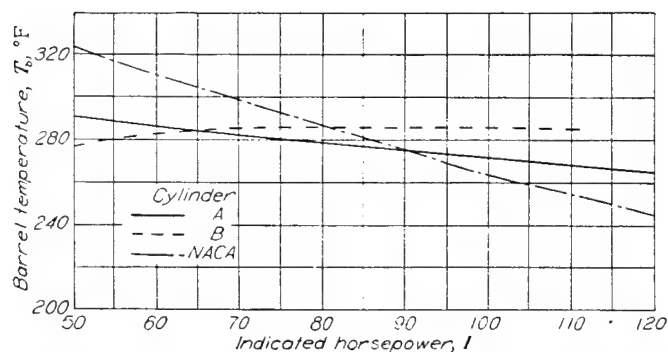
FIGURE 16.—Barrel temperature of A, B, and NACA cylinders for inside-wall temperature of 500° F. Cooling-air pressure at inlet, 29.92 inches Hg absolute; cooling-air temperature, 100° F; fuel-air ratio, 0.08.

of these cylinders lower than those of the B cylinder over part of the range.

Comparison of outside-wall heat-transfer coefficients.— Calculations were made of the outside-wall heat-transfer coefficients U for the NACA cylinder and the results were compared with coefficients for the A and B cylinders. The outside-wall coefficients for the head and the barrel of a cylinder can be calculated from the equation

$$U = K(\Delta p \rho / \rho_{70})^m$$

The values of the constants K and m for the head and the barrel were obtained from equations (1) and (2), respectively.

The results of the calculations are shown in figure 17. At pressure drops above 9 inches of water, the heat-transfer coefficients of the head of the NACA cylinder are less than those of the B cylinder; above 30 inches of water, they are less than those of the A cylinder. This result was entirely unexpected because of the better fin proportions of the NACA cylinder. When a perfect thermal bond was assumed for the NACA cylinder, the estimated heat-transfer coefficients
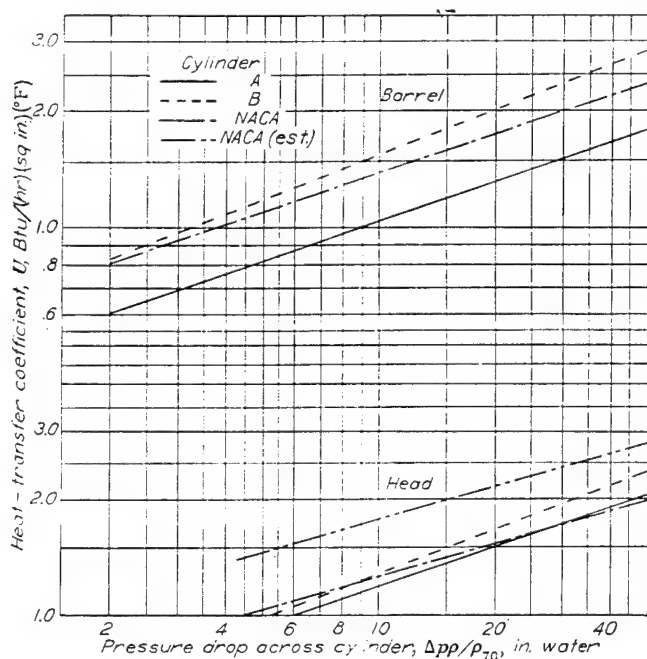

FIGURE 17.—Variation of outside-wall heat-transfer coefficients with pressure drop across cylinder for A, B, and NACA cylinders. Estimated NACA coefficient based on thermally perfect bond.
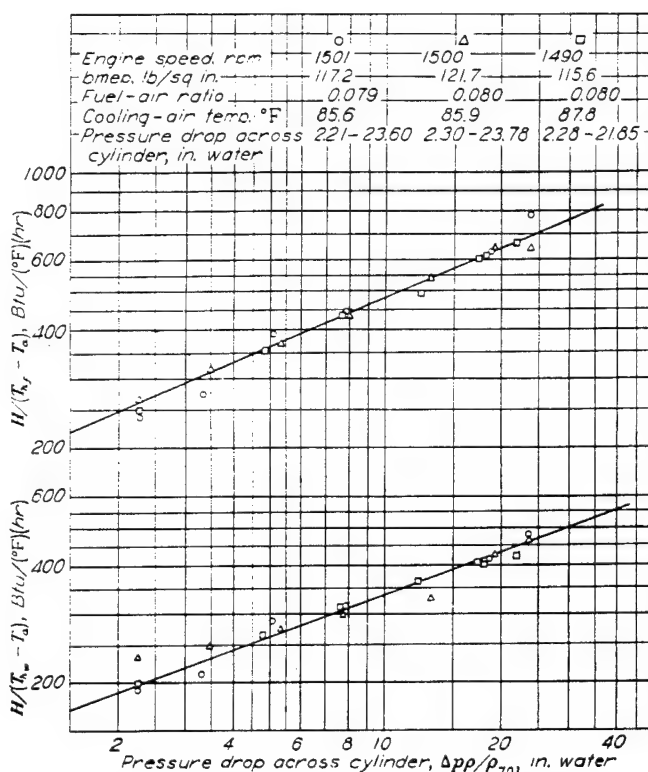

FIGURE 18.—Effect of pressure drop across cylinder on $H/(T_{h,w} - T_a)$ and $H/(T_{h,f} - T_a)$ for NACA cylinder. No boost; spark timing, 27° B.T.C.; carburetor-air temperature, 77° F.

showed considerable improvement over those of the B cylinder. These coefficients are shown in figure 17 by the broken line.

Another interesting result of figure 17 is that the heat-transfer coefficients of the barrel of the NACA cylinder, which had steel fins, were slightly less for most pressure drops than the coefficients of the barrel of the B cylinder, which had an aluminum muff with aluminum fins over the steel barrel liner. From data on electrically heated cylinder barrels (references 2 and 7) the heat-transfer coefficients for barrels of the B and the NACA cylinders were computed. The calculations, which were made to check the results obtained from single-cylinder tests, showed that the coefficients for a barrel with fin proportions and material similar to those of the B cylinder were higher than those for a barrel with fin proportions and material similar to those of the NACA cylinder. The calculations therefore checked the trend of the barrel curves of figure 17.

The outside-wall heat-transfer coefficients of the head of the NACA cylinder were based on the difference between the wall temperatures and the cooling-air temperatures. Data were obtained on the temperatures of the fins adjacent to the wall of the head. The heat dissipated per degree Fahrenheit difference between the wall temperatures and the cooling-air temperature, and the heat dissipated per degree Fahrenheit difference between the fin temperatures adjacent to the walls and the cooling-air temperature were therefore calculated for the head of the NACA cylinder from the single-cylinder engine tests. These heat values are proportional to the heat-transfer coefficients. The results of the calculations are shown in figure 18. The temperature $T_{h,f}$ is the
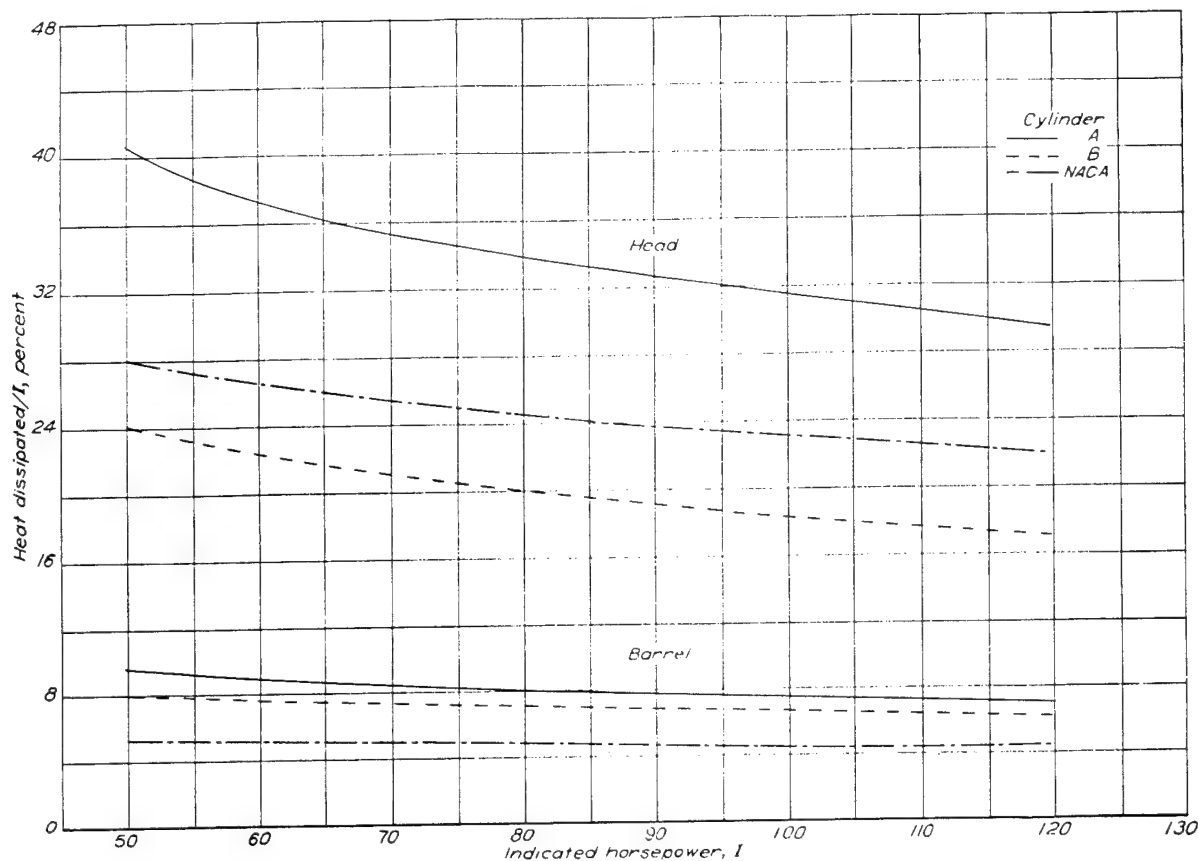
FIGURE 19.—Heat dissipated to cooling air passing over heads and barrels of A, B, and NACA cylinders.

average of the 10 thermocouples, identified by the suffix A, installed on the root of the fins; the temperature $T_{h,w}$ is the average of the same numbered thermocouples installed on the head. The heat dissipated per degree Fahrenheit difference between fin and cooling-air temperatures is about 40 percent greater than a corresponding heat transfer based on wall temperature. Thus, the bond between the fins and the head of the NACA cylinder was not thermally perfect; an appreciable temperature drop from wall to fins was obtained. If the thermal bond between the fins and the wall were thermally perfect, the outside-wall coefficients of the NACA cylinder head would be about 40 percent greater than those shown and would be of the order expected.

Comparison of heat dissipation.—Of interest in the design of air-cooled cylinders is the percentage of indicated horsepower dissipated as heat to the cooling air. Calculations have been made for the barrels and the heads of the three cylinders of the percentage heat dissipated to the cooling air at various indicated horsepowers under the same cooling conditions used in figure 13. The results are shown in figure 19. With an indicated horsepower of 120, the percentage of heat dissipated from the B cylinder head is approximately 17 percent; from the A cylinder head, about 29.5 percent; from the NACA cylinder head, 22 percent. From values given in figure 19, coefficients of heat transfer can be estimated for given horsepower outputs, cylinder temperatures, and cooling-air temperatures in the design of new cylinders.

Comparison of power required for cooling.—The percentage power required for the three cylinders for various

indicated horsepowers and with a rear spark-plug temperature of 450° F is shown in figure 20. The power required for cooling was obtained from the relation

$$hp = \frac{\text{weight of air} \times \text{pressure drop}}{\rho_1 g \times 33,000}$$

in which the weight of air is in pounds per minute, the pressure drop is in pounds per square foot, and $\rho_1 g$ is in pounds per cubic foot. The pressure drop for cooling the NACA cylinder was calculated from equation (1) and the weight of air was obtained from figure 8. The head temperature corresponding to the rear spark-plug temperature of 450° F was obtained from an extrapolated curve of figure 11. The specific weight of air $\rho_1 g$ was based on Army sea-level pressure of 29.92 inches of mercury absolute and temperature of 100° F. The powers for the A and the B cylinders were calculated from similar data. The calculated powers are the internal-drag horsepowers across the cylinders and should not be confused with the power required for cooling engines placed in moving ducts or nacelles.

The percentage of the indicated horsepower required for cooling the NACA and the B cylinders is very low, about 0.8 percent at an indicated horsepower of 120; but the A cylinder would have required about 28 percent if such a power could have been developed by this cylinder. These values illustrate the great improvement in cylinder cooling that has resulted from better fin proportions. For the greater part of the range of indicated powers shown in figure 20, the NACA cylinder shows a little less power required for cooling

than the B cylinder. Over the entire range of powers shown, the NACA cylinder curve should be much lower than the B cylinder curve; the high percentages for the NACA cylinder curve are due to the poor thermal bond between the fins and the wall of the NACA cylinder head.

Calculations similar to those of figure 20 were also made for a constant inside-wall head temperature of 500° F instead of a constant rear spark-plug temperature of 450° F. The pressure drop required for cooling was obtained by using equation (7); constants in the equation for the NACA cylinder were obtained from the present data; for the A cylinder, from reference 1; and for the B cylinder, from unpublished data on that cylinder. The results of the calculations are given in figure 21. Owing to an imperfect thermal bond between the fins and the wall of the NACA cylinder, above
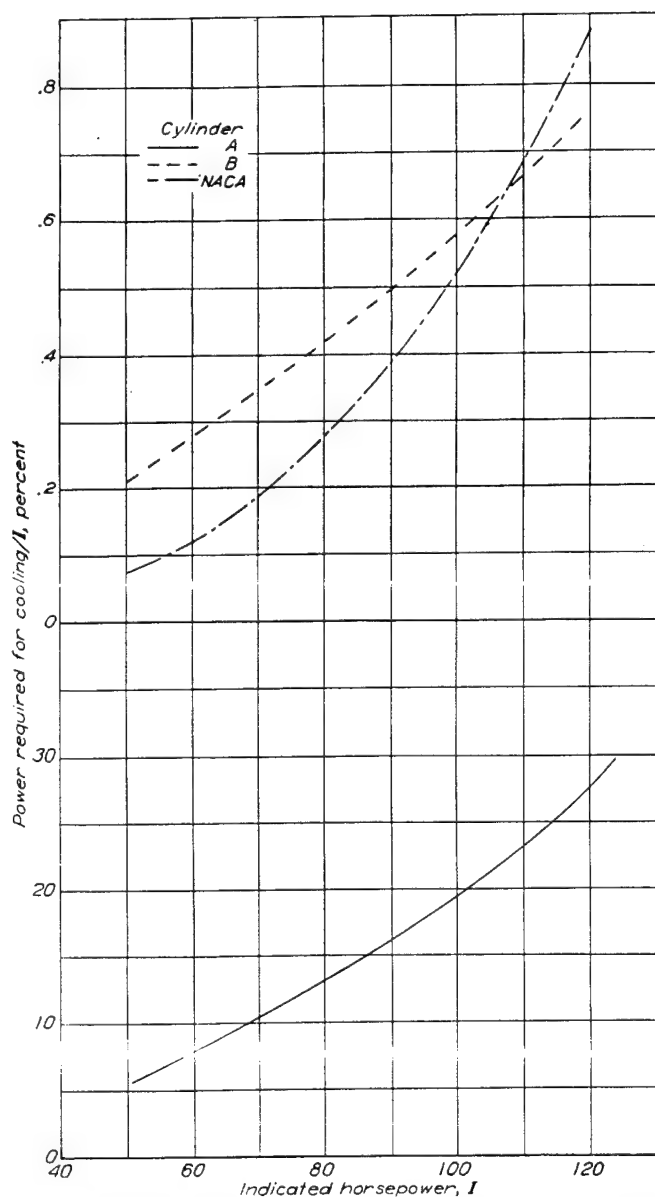

FIGURE 20.—Percentage indicated horsepower required for cooling A, B, and NACA cylinders. Cooling-air pressure at inlet, 29.92 inches Hg absolute; cooling-air temperature, 100° F; fuel-air ratio, 0.08; rear spark-plug temperature, 450° F.
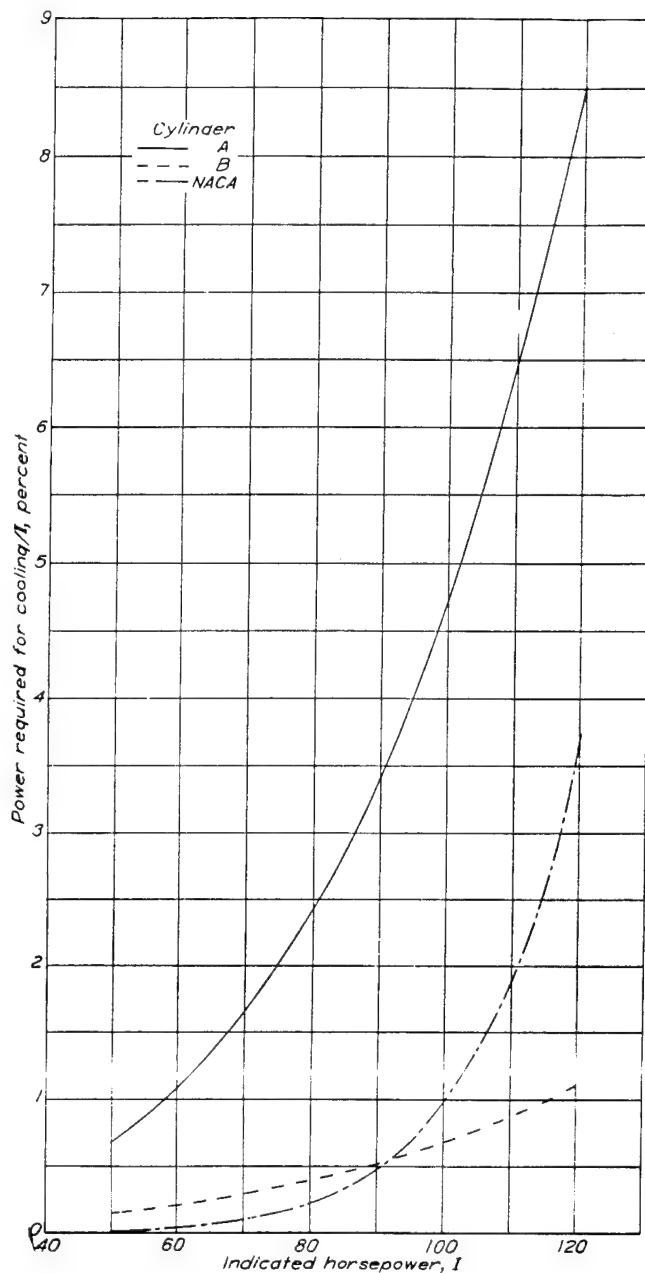

FIGURE 21.—Percentage indicated horsepower required for cooling A, B, and NACA cylinders. Cooling-air pressure at inlet, 29.92 inches Hg absolute; cooling-air temperature 100° F; fuel-air ratio, 0.08; inside-wall temperature of head, 500° F.

an indicated horsepower of 90 the power required for cooling the NACA cylinder is greater than the power required for cooling the B cylinder. Thus, at an indicated horsepower of 120, about 1 percent of the indicated horsepower is required for cooling of the B cylinder and almost 4 percent for the NACA cylinder. Because the average inside-wall temperature of the A cylinder head was much less than 500° F for a rear spark-plug temperature of 450° F, much less cooling power (about 8.5 percent) is required to hold an inside-wall temperature of 500° F than to hold a rear spark-plug temperature of 450° F (about 28 percent) at an indicated horsepower of 120. (See figs. 13, 14, and 15.)

ESTIMATED EFFECT OF INCREASED COOLING SURFACE AND CYLINDER-WALL THICKNESS ON ENGINE PERFORMANCE

The performance of the NACA cylinder has been compared with the performance of two commercial cylinders. The three cylinders differed in head thickness, head fin-surface area, barrel finning, and barrel material. It is difficult under these conditions to determine the exact effect on performance of increasing the head fin-surface area and of varying the head-wall thickness. Calculations have therefore been made of the estimated pressure drop required to cool at constant power output hypothetical cylinders with a perfect thermal bond between the wall and the fins, with varying head-wall thickness, and with two fin-surface areas. In addition, calculations of the estimated power output for constant cooling pressure drop were made for the cylinders.

As was previously stated, the assumption has been made in these calculations that the thermal bond between the fins and the wall was perfect for all the hypothetical cylinders. It is possible to obtain a perfect thermal bond between the fins and the wall when preformed fins are used. A steel cylinder barrel with aluminum fins, which was manufactured later than the NACA cylinder, has been tested at LMAL. The fins were preformed and cast in an aluminum base by a method similar to that used on the NACA cylinder; the aluminum base, in turn, was bonded to the steel barrel. A sketch of a cross section of the cylinder is included in figure 22. Thermocouples were placed on the aluminum base at the locations shown in figure 22, the cylinder was electrically heated, and the heat loss was determined for various amounts of cooling air passing over the cylinder. The calculated over-all heat-transfer coefficients U for a perfect thermal bond between the fin and the wall are shown for various weight flows of cooling air by the curve in figure 22. The experimental coefficients based on the wall temperatures are shown by the points. The agreement between the calculated and the experimental values is very good, which indicates

that this test cylinder has a thermal bond between the fins and the wall almost equivalent to the thermal bond of an integral fin-cylinder wall combination.

An equation based on inside-wall temperature has been developed in the appendix as equation (16) for the pressure drop required for cooling for given cooling and power conditions, inside-wall and outside-wall areas, thickness of the cylinder wall, and cylinder material.

$$\Delta p = \frac{\rho_{70}}{\rho} \left[\frac{\overline{A} I^x a_1 k_m (T_g - T_{h,1})}{K a_0 k_m (T_{h,1} - T_a) - K a_0 t_w \overline{A} I^x (T_g - T_{h,1})} \right]^{1/m} \quad (16)$$

where $\overline{A}$ and x are constants for selected ranges of I as explained in the appendix. By means of this equation the curves of figures 23 and 24 were obtained. The calculations were made for hypothetical cylinders with the following characteristics:

Cylinder group	1	2	3	4
Inside-head proportions	B cylinder		NACA cylinder	
Head-fin proportions	B cylinder	NACA cylinder	B cylinder	NACA cylinder
Head thickness, inches	Varying from 0.75 to 1.50		Varying from 0.75 to 1.50	
Barrel	B cylinder		NACA cylinder	

Group 1 has been compared with group 2, and group 3 with group 4. Comparisons made in this way were thought to be fairer than comparisons of hypothetical cylinders basically similar to the B cylinder (group 1) with hypothetical cylinders basically similar to the NACA cylinder (group 4). The values of the constants K and m in equation (16) for cylinder groups 1 and 3 were obtained from unpublished test data on the B cylinder. For groups 2 and 4 the constant K in equation (16) was obtained by increasing the
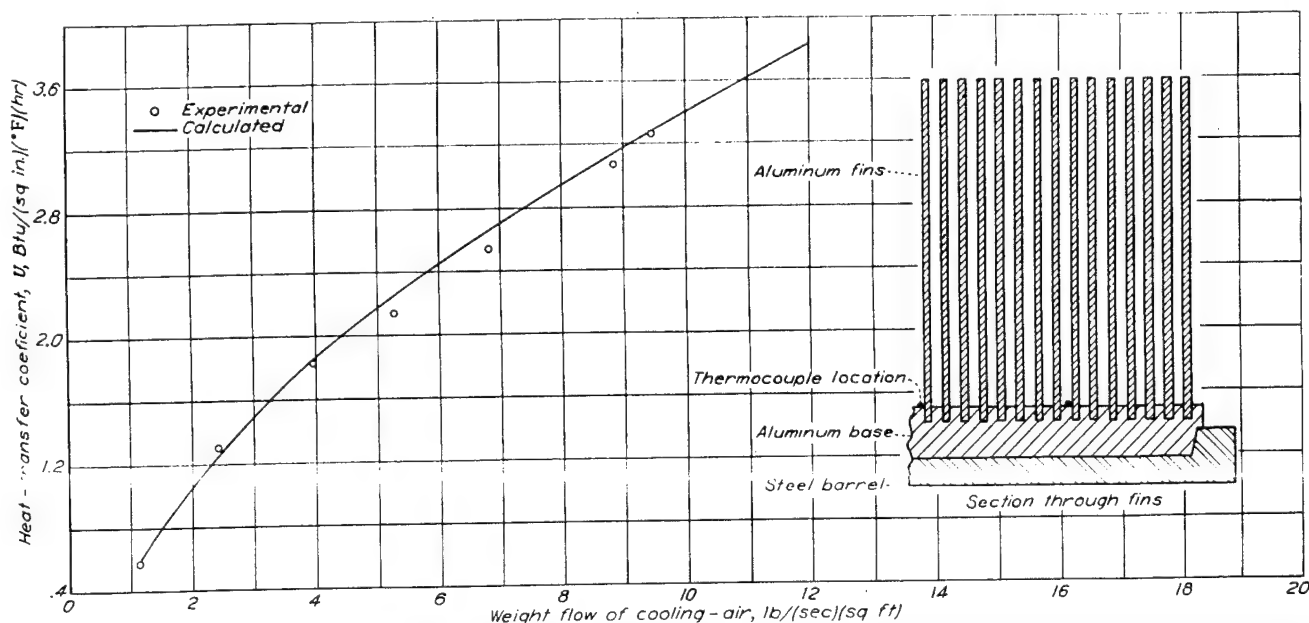
FIGURE 22.—Comparison of experimental over-all heat-transfer coefficients with calculated coefficients. Experimental coefficients for test cylinder based on temperature difference between aluminum base and cooling air. Calculated coefficients for aluminum cylinder with aluminum fins based on thermal conductivity $k_m = 9.22$ Btu/(hr)(sq in.) (°F/in.). Cylinder diameter, 6.34 inches; fin width, 1.28 inches; fin thickness, 0.031 inch; fin space, 0.045 inch.
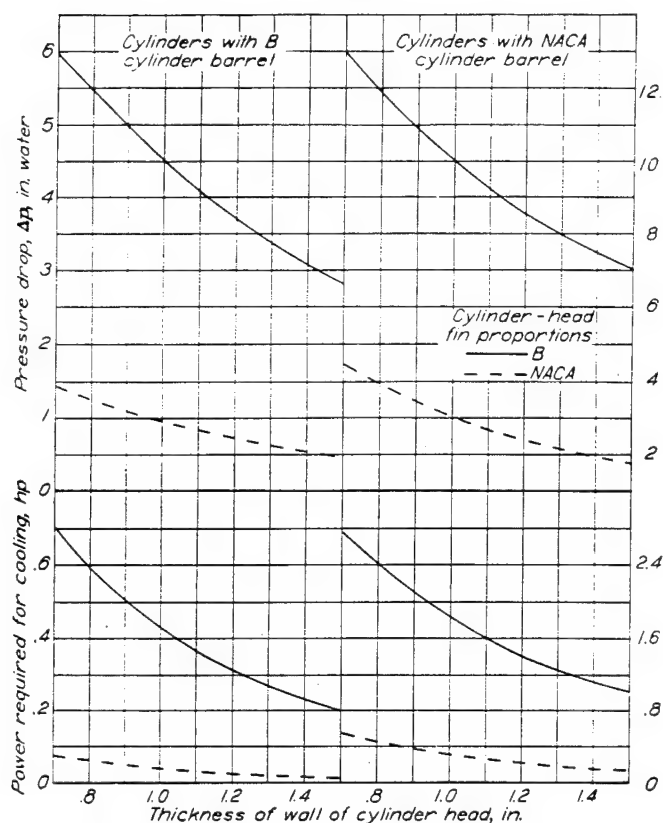
FIGURE 23.—Effect of wall thickness on pressure drop and power required for cooling with B and NACA cylinder-head fin proportions at cruising power.   Fuel-air ratio, 0.08; cooling-air temperature, 100° F; cooling-air pressure at inlet, 29.92 inches Hg absolute; inside-wall temperaure, 500° F; indicated horsepower per cylinder, 67.8.

constant K in equation (1) by 40 percent because the heat-transfer coefficient for the NACA cylinder head would be increased 40 percent by a perfect thermal bond.   (See fig. 18.) The constant m in equation (1) was used for groups 2 and 4.

The values of $\overline{A}$ and x were calculated from equations (10) and (15) of the appendix; the data needed for the calculations were obtained for groups 1 and 2 from unpublished test data on the B cylinder and for groups 3 and 4, from data on the NACA cylinder.   The constant $\overline{A}$ and the exponent x in equation (16) are independent of wall thickness and type of finning.   The value of T_g for all groups was obtained from a curve of T_g plotted against fuel-air ratio.   A value of a_1 of 78.4 square inches was used for groups 1 and 2 and 76.8 square inches, for groups 3 and 4.   (See table I.)   The outside-wall areas a_0 were estimated for groups 1 and 2 from the thickness and the outside-wall area of the B cylinder and for groups 3 and 4, from those of the NACA cylinder. The calculations were based on Army sea-level pressure and temperature and on a constant inside-wall head temperature of 500° F.   The density ρ was assumed to be inlet density. The pressure drops calculated by means of equation (16) and the constants given are equivalent to the weight of air required to cool the cylinders for given conditions and do not correspond to the pressure drops that would exist for these air weights on the hypothetical cylinders because of the changing path lengths.   The pressure drops obtained from equation (16) were therefore corrected for the path length of each cylinder.

Figure 23 shows the results of calculations for the four groups of cylinders for an indicated horsepower of 67.8 per cylinder, approximately cruising power, and a fuel-air ratio of 0.08.   A large decrease in pressure drop, an average of approximately 80 percent over the range of wall thickness shown, can be obtained by using head fins of NACA cylinder proportions rather than fins of B cylinder proportions.   Over the range of wall thickness shown, the required pressure drop decreased with all cylinder groups as the thickness increased. In effect, the increase in heat transfer caused by the increase in fin surface as the wall thickness increased more than balanced the decrease in heat transfer caused by the thicker walls.   A maximum thickness will be reached beyond which the pressure drop will start increasing.   The decrease in pressure drop was approximately 50 percent (from initial pressure drops of 3.66 and 5.21 in. of water) for the cylinders with the B cylinder-head fin proportions; and 75 percent (from initial pressure drops of 0.74 and 1.23 in. of water) for the cylinders with NACA cylinder-head fin proportions.

The powers required for cooling the four cylinder groups are also plotted in figure 23.   The average decrease in cooling power over the range of wall thicknesses is shown more than 90 percent when NACA cylinder fin proportions are used instead of the B cylinder fin proportions.   The decrease in cooling power as wall thickness increases is also large for all cylinder groups.   In no case, however, is the cooling power greater than 1 percent of the indicated power.

The results of calculations for an indicated horsepower of 133.3 per cylinder, approximately take-off power, and a fuel-air ratio of 0.10 for the four groups of cylinders are shown in figure 24.   A large decrease in pressure drop required for cooling is again obtained by using NACA cylinder fin proportions instead of the B cylinder fin proportions on the heads of the hypothetical cylinders; this decrease is approximately 80 percent for cylinder groups 1 and 2 and about 70 percent for cylinder groups 3 and 4.   For the B cylinder-head fin proportions, a reduction in pressure drop from 6 to 2.8 inches of water in group 1 and from 13 to 7 inches of water in group 3 is obtained by increasing the head-wall thickness from 0.7 inch to 1.5 inches.   The corresponding decreases in power required for cooling, also shown in figure 24, are 92 and 84 percent, respectively.   Although these reductions in cooling power are large, the fact that the maximum cooling power is only 2 percent of the indicated horsepower makes them less important than the reduction in cooling-pressure drop.   The pressure drop required for cooling in the four groups decreases as the thickness of the cylinder-head wall increases for the range of thickness shown; the decrease based on the pressure drop for a wall thickness of 0.7 inch is about 50 percent for the cylinders with B cylinder fin proportions and 65 percent for the cylinders with NACA cylinder fin proportions.   These percentage decreases of pressure drop due to the increase of wall thickness and to the changing of fin proportions to obtain large fin-surface areas are of the order of pressure-drop decreases obtained in figure 23 for the cruising-power condition.

Calculations for the four groups have also been made of the estimated power output that could be obtained for a pressure drop of 7 inches of water across the cylinder, a
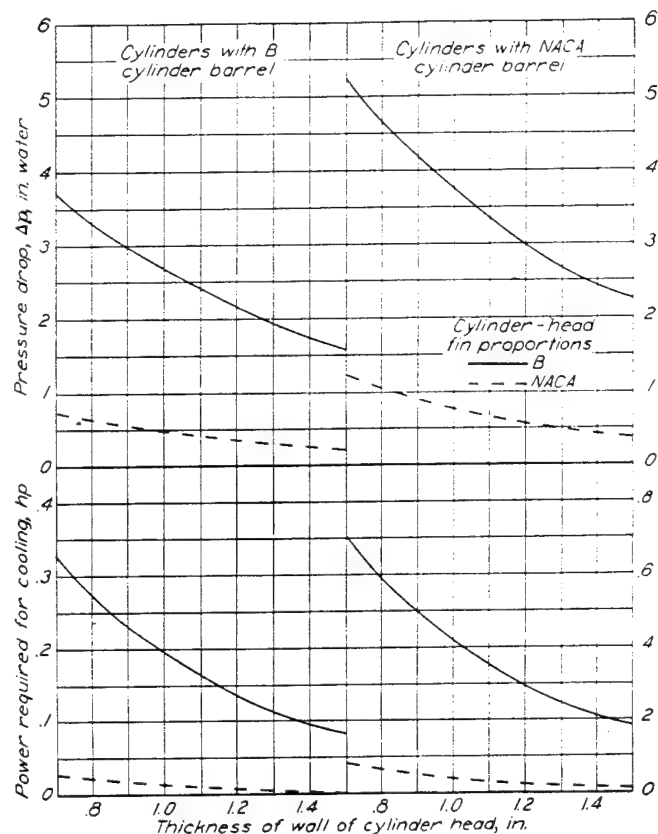
FIGURE 24.—Effect of wall thickness on pressure drop and power required for cooling with B and NACA cylinder-head fin proportions at take-off power. Fuel-air ratio, 0.10; cooling-air temperature, 100° F; cooling-air pressure at inlet, 29.92 inches Hg absolute; inside-wall-temperature, 500° F; indicated horsepower per cylinder, 133.3.

constant inside-wall head temperature of 500° F, and a fuel-air ratio of 0.10 for a range of head-wall thicknesses from 0.7 inch to 1.9 inches. The calculations were based on Army standard sea-level pressure and temperature. Equation (17) of the appendix was used for calculating the power outputs.

$$I = \left\{ \frac{K a_0 (\Delta p \rho / \rho_{70})^m k_m (T_{h,1} - T_a)}{\overline{A}(T_g - T_{h,1})[a_1 k_m + K a_0 (\Delta p \rho / \rho_{70})^m t_w]} \right\}^{1/x} \qquad (17)$$

The constants in the equations for the four groups of cylinders are the same used in equations (8), (10), (15), and (17) for determining the results of figures 23 and 24. The method of determining $\overline{A}$ and x is given in the appendix. In order to use, in the calculations for figure 25, the constants K and m of equation (16), the pressure drop of 7 inches of water had to be corrected for flow-path length of the cylinders to obtain pressure drops equivalent to the weight of air actually flowing across the cylinders.

The results of the calculations are shown in figure 25. The curves for the cylinders with NACA cylinder fin proportions on the heads show a maximum estimated power output at a wall thickness of about 1.6 inches; the curves for cylinders with B cylinder fin proportions on the heads show a maximum output at a wall thickness of about 1.8 inches. Above such thicknesses the loss in heat transfer due to the thick wall more than balances the gain in heat transfer due to the large outside-wall areas of the heads with resultant large fin-surface areas. The gain in power output in changing from a wall thickness of 0.7 inch to optimum wall thickness is from 10 to 20 percent for cylinder groups 2, 3, and 4 and about 30 percent for cylinder group 1. A gain of about 35 percent in maximum power output is possible by using NACA cylinder fin proportions instead of B cylinder fin proportions for the cylinders with B cylinder barrel, and 24 percent for the cylinders with the NACA cylinder barrel. By the use of optimum fin proportions on the heads of the cylinders instead of NACA cylinder proportions, the value of 35 percent could be increased to approximately 50 percent. Figure 25 shows that power outputs of 260 indicated horsepower (about 4700 ihp for 18 cylinders) could be obtained with a cylinder of 1.6 inches head-wall thickness, NACA cylinder fin proportions on the head, and a B cylinder barrel. A power output of 260 indicated horsepower illustrates the possibilities of the air-cooled engine solely on the basis of cooling considerations. The attainment of such powers in future engines will depend not only on a change in the design of the cooling surfaces but also on mechanical modifications of the engine and the improvement of fuels and spark plugs. An indicated mean
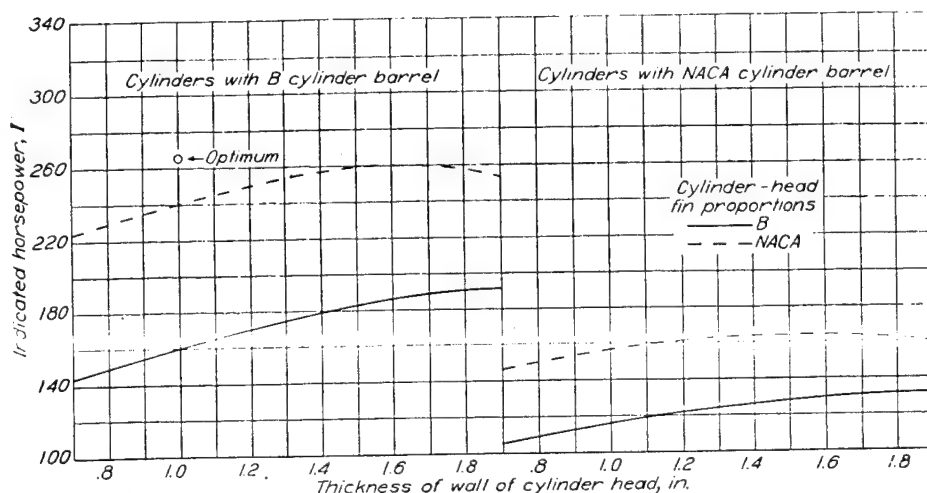

FIGURE 25.—Estimated power for various cylinder-head wall thicknesses for cylinders with fin proportions of the B and the NACA cylinder heads. Fuel-air ratio, 0.10; cooling-air temperature, 100° F; cooling-air pressure, 29.92 inches Hg absolute; inside-wall temperature, 500° F; pressure drop for cooling, 7 inches water.

effective pressure of 288 pounds per square inch was obtained in the tests of the NACA cylinder, which is slightly higher than present values of take-off indicated mean effective pressures. No tests were made at a higher power output because the hold-down bolts in the flange of the cylinder could not withstand the higher loads.

The increase of cylinder-wall thickness and the addition of fin surface to a cylinder, with the consequent addition of cylinder weight to an engine, will depend upon the service requirements of the engine. The increased weight for adequate cooling of the cylinder would not be advisable if the high power output made possible by such an addition could not be obtained because of knock or preignition characteristics of the fuel. In addition, if a high power is necessary only for take-off, the addition of cylinder weight for adequate cooling for such a short period would be uneconomical. It would probably be advisable to cool the engine for such a short period by the use of an overrich mixture.

CONCLUSIONS

1. The tests of the NACA cylinder showed that the outside-wall heat-transfer coefficient could be increased 40 percent over that obtained on the present NACA cylinder by providing a perfect thermal bond between the fins and the wall of the head.

2. Estimates showed that, for a range of power from present cruising to take-off power, and for a range of wall thicknesses from 0.7 inch to 1.5 inches, the pressure drop required to cool a commercial cylinder could be decreased as much as 80 percent if the head fins are replaced by fins of NACA cylinder proportions and if the thermal bond between the walls and the proposed fins is perfect.

3. On a basis of a constant inside-wall head temperature of 500° F, it is estimated that the substitution of NACA cylinder fin proportions for the fins on the heads of commercial cylinders will permit an increase in power output of approximately 35 percent. If optimum fin proportions are used, this value is increased to 50 percent. These estimated increases are solely on the basis of cooling and do not take into account any of the other factors limiting power output.

4. As cylinder-wall thickness increases, the heat transfer decreases because of the greater path length for the heat flow and the heat transfer increases because of the greater outside-wall area; thus, fin-surface area increases if the fin proportions are held constant. Calculations of estimated power outputs of hypothetical cylinders show that the optimum thickness for heat transfer is approximately 1.7 inches.

5. Calculations based on a constant inside-wall head temperature of 500° F indicated that the pressure drop required for cooling hypothetical cylinders of varying head-wall thickness could be decreased 50 percent by increasing the thickness from 0.7 inch to 1.5 inches.

6. Based only on cooling considerations, an estimated output of 260 indicated horsepower could be developed by a hypothetical cylinder with an aluminum muff on the barrel, NACA cylinder-head fin proportions, a wall thickness of 1.6 inches, 7 inches of water pressure drop available for cooling, and a constant inside-wall head temperature of 500° F.

APPENDIX

DEVELOPMENT OF EQUATIONS FOR ESTIMATED POWER OUTPUT AND PRESSURE DROP REQUIRED FOR COOLING

The rate of heat transfer (Btu/hr) from the combustion gas to the cylinder head in the range of fuel-air ratios to the rich side of the theoretically correct mixture may be written as a good first approximation (reference 1),

$$H = \bar{B} a_1 I^{n'} (T_g - T_h) \tag{3}$$

For the rate of heat transfer from the cylinder head to the cooling air, the following equation is given in reference 1:

$$H_1 = K a_0 (\Delta p \rho / \rho_{70})^m (T_h - T_a) \tag{4}$$

The conduction of heat through the head may be expressed by the following equation (reference 6):

$$H = a_1 \frac{k_m}{t_w} (T_{h,1} - T_h) \tag{5}$$

In the foregoing equations

H	heat transferred per unit time from combustion gas to cylinder head, Btu/hr
$\bar{B}$ and K	constants
a_1	internal area of head of cylinder, sq in.
I	indicated horsepower per cylinder
m and n'	constants
T_g	effective gas temperature, °F
T_h	average temperature over outside cylinder-head surface when equilibrium is attained, °F
T_a	inlet temperature of cooling air, °F
H_1	heat transferred per unit time from cylinder head to cooling air, Btu/hr
a_0	outside-wall area of head of cylinder, sq in.
Δp	pressure drop across cylinder including loss from jacket exit, in. water
ρ	average density of cooling air entering and leaving fins, lb ft^{-4} sec^2
ρ_{70}	density of air at 29.92 inches of mercury and 70° F, lb ft^{-4} sec^2
k_m	thermal conductivity of head, Btu/(hr) (sq in.) (° F/ in.)
t_w	average thickness of wall, in.
$T_{h,1}$	average inside-head temperature, °F

For equilibrium $H = H_1$

Equations (3) and (5) can be combined to obtain

$$T_{h,1} = \frac{\bar{B} a_1 I^{n'} (T_g - T_h)}{a_1 k_m / t_w} + T_h \tag{6}$$

Equations (3), (4), and (5) can be combined and T_h can be eliminated to give Δp as a function of $T_{h,1}$

$$\Delta p = \frac{\rho_{70}}{\rho} \left\{ \frac{\bar{B} a_1 I^{n'} k_m (T_{h,1} - T_g)}{K a_0 [\bar{B} I^{n'} t_w (T_g - T_a) - k_m (T_{h,1} - T_a)]} \right\}^{1/m} \tag{7}$$

The mean coefficient over the entire cycle for the transfer of heat from the combustion gas to the cylinder is defined in reference 1 as

$$q_0 = \frac{H}{a_1 (T_g - T_h)} \tag{8}$$

When equation (8) is compared with equation (3), the mean coefficient is equal to $\bar{B} I^{n'}$. Because q_0 is based on outside-wall temperature, it is an over-all heat-transfer coefficient and takes into account the transfer of heat by convection from the gas to the inner wall and the transfer of heat by conduction from the inner to the outer wall. The heat-transfer coefficient q_1 for convection of heat from the gas to the inner wall is written

$$q_1 = \frac{H}{a_1 (T_g - T_{h,1})} \tag{9}$$

When equations (5), (8), and (9) are combined, the coefficient q_1 in terms of the over-all coefficient q_0 may be obtained or,

$$q_1 = \frac{q_0}{1 - \frac{t_w}{k_m} q_0} \tag{10}$$

A combination of equations (4) and (9) gives

$$\frac{T_g - T_{h,1}}{T_h - T_a} = \frac{K a_0 (\Delta p \rho / \rho_{70})^m}{q_1 a_1} \tag{11}$$

A combination of equations (4) and (5) gives

$$\frac{T_{h,1} - T_h}{T_h - T_a} = \frac{K a_0 (\Delta p \rho / \rho_{70})^m}{a_1 k_m / t_w} \tag{12}$$

Solving equations (11) and (12) for T_h and equating the resulting equations give

$$\frac{q_1 a_1 (T_g - T_{h,1}) + K a_0 (\Delta p \rho / \rho_{70})^m T_a}{K a_0 (\Delta p \rho / \rho_{70})^m} = \frac{a_1 k_m T_{h,1} + K a_0 (\Delta p \rho / \rho_{70})^m t_w T_a}{a_1 k_m + K a_0 (\Delta p \rho / \rho_{70})^m t_w} \tag{13}$$

Because q_1 is the heat-transfer coefficient for the convection of heat from the gas to the inner wall, this coefficient is dependent only on I; whereas the coefficient for the convection of heat from the gas to the outer wall q_0 is dependent on both I and t_w. For a particular cylinder the variation

of q_1 with I can be obtained by substituting the value of $\overline{B}I^{n'}$ for q_0 in equation (10); the values of $\overline{B}$ and n' are determined from tests of the cylinder. When the values of t_w and k_m for the cylinder are also substituted in equation (10), the variation of q_1 with I becomes

$$q_1 = \frac{\overline{B}I^{n'}}{1 - \frac{t_w}{k_m}\overline{B}I^{n'}} \tag{14}$$

For limited ranges of I this variation can be approximated by

$$q_1 = \overline{A}I^z \tag{15}$$

where $\overline{A}$ is a constant. In selected ranges of I the error involved in the approximation is less than 1 percent, and values for $\overline{A}$ and n can be determined for each range. Equation (15) is valid for any cylinder with the same inside dimensions and combustion conditions as the cylinder from which the constants are obtained.

Substitution of equation (15) in equation (13) gives a solution for Δp:

$$\Delta p = \frac{\rho_{70}}{\rho}\left[\frac{\overline{A}I^z a_1 k_m(T_g - T_{h,1})}{Ka_0 k_m(T_{h,1} - T_a) - Ka_0 t_w \overline{A}I^z(T_g - T_{h,1})}\right]^{1/m} \tag{16}$$

In like manner equation (13) can be solved for I to obtain

$$I = \left\{\frac{Ka_0(\Delta p\rho/\rho_{70})^m k_m(T_{h,1} - T_a)}{\overline{A}(T_g - T_{h,1})[a_1 k_m + Ka_0(\Delta p\rho/\rho_{70})^m t_w]}\right\}^{1/z} \tag{17}$$

The pressure drop required for cooling for various wall thicknesses, and thus various outside-wall areas, can be obtained from equation (16) for given values of I, $T_{h,1}$, and T_a. The power that can be developed for various wall thicknesses can be obtained from equation (17) for given values of $\Delta p\rho/\rho_{70}$, $T_{h,1}$, and T_a.

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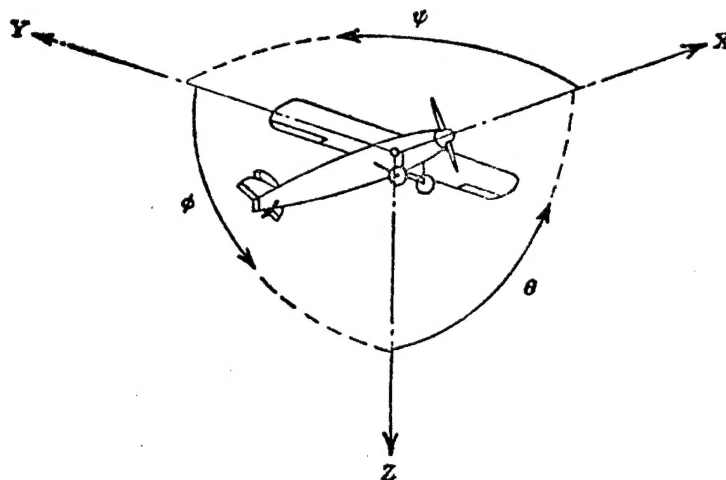
Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Sym-bol		Designation	Sym-bol	Positive direction	Designa-tion	Sym-bol	Linear (compo-nent along axis)	Angular
Longitudinal	X	X	Rolling	L	$Y \longrightarrow Z$	Roll	φ	u	p
Lateral	Y	Y	Pitching	M	$Z \longrightarrow X$	Pitch	θ	v	q
Normal	Z	Z	Yawing	N	$X \longrightarrow Y$	Yaw	ψ	w	r

Absolute coefficients of moment

$$C_l = \frac{L}{qbS} \qquad C_m = \frac{M}{qcS} \qquad C_n = \frac{N}{qbS}$$
$$\text{(rolling)} \qquad \text{(pitching)} \qquad \text{(yawing)}$$

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D Diameter
p Geometric pitch
p/D Pitch ratio
V' Inflow velocity
V_s Slipstream velocity
T Thrust, absolute coefficient $C_T = \dfrac{T}{\rho n^2 D^4}$

Q Torque, absolute coefficient $C_Q = \dfrac{Q}{\rho n^2 D^5}$

P Power, absolute coefficient $C_P = \dfrac{P}{\rho n^3 D^5}$

C_s Speed-power coefficient $= \sqrt[5]{\dfrac{\rho V^5}{P n^2}}$

η Efficiency
n Revolutions per second, rps

Φ Effective helix angle $= \tan^{-1}\left(\dfrac{V}{2\pi rn}\right)$

5. NUMERICAL RELATIONS

1 hp=76.04 kg-m/s=550 ft-lb/sec
1 metric horsepower=0.9863 hp
1 mph=0.4470 mps
1 mps=2.2369 mph

1 lb=0.4536 kg
1 kg=2.2046 lb
1 mi=1,609.35 m=5,280 ft
1 m=3.2808 ft